PRODUCT ENGINEERING

KENNETH H. CONDIT
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Oppose Plans to Limit Equipment Buying

RECENT events in Washington point to a new engineering responsibility—to assail the type of company management represented by code draft agreements not to purchase production equipment, and not to bring out new models. Association after association has come to Washington with rigid agreements not to buy any equipment for two years. Agreements of machinery groups not to bring out new models are scarcer but equally reprehensible.

Strange as it may seem to those of us who recognize the essential part played by engineering development in our present mode of life, the very men who should be the most ardent supporters of continuing progress are conspiring to block it. Their action is the harder to understand because they have made fortunes through the development they now seek to obstruct.

The engineer has a vital personal interest in fighting this reactionary tendency on the part of industrial executives, but it is also his duty to the community, as an educated man, to raise his voice against action that will go far to counteract any good that may come from NRA effort.

In his own company and toward his employer the engineer may have to act with a certain amount of diplomacy. It is not necessary, however, to spit in a man's eye to get his attention. Constant insistence on the soundness of a principle carries conviction in the long run, particularly if the adversary has no sound arguments to support his course.

As for his customers who harbor this mistaken notion, they should be labored with on every occasion. The sales force and the advertising copy of the company will be the front line troops in the attack, but the engineer can do much to support them by developing logical reasons for their stand, and by making the most of his contacts through society meetings and plant visits. He has a reponsibility that cannot be avoided, and that he will not wish to avoid.

Engineering Is Not a Game

ONE of the things that definitely stamps the quality of the designing engineer is the manner in which he handles rules and empirical formulas. There are mechanical designers (emphasis on the word mechanical) who religiously adhere to the dictates of handbooks. Often they classify themselves as conservatives. Possibly this is to cover an inferiority complex. The fruits of their labors are too often clumsy in appearance and lack the refinements that are the trademark of a master hand.

Empirical design requires that the pulley diameter be fifteen or more times greater than the belt thickness. But in some conveyor belts the surface wear is extremely great. Why not make the pulley smaller? Similarly, metals in sliding contact should always be dissimilar, but cast-iron piston rings in cast-iron cylinder blocks are considered by many to be the best obtainable combination. A hollow shaft is stronger, but calculations will show that if the twisting loads are suddenly applied a solid shaft is more economical. Many similar instances could be cited.

There are no inflexible rules in design. There cannot be because engineering is a live and growing profession. Old conceptions change, new discoveries are made. Final designs must often be compromises between two opposing ideals. To establish ideals requires knowledge and imagination. It is not a game; there are no fast rules.

Fig. 1—In the design of this light-weight rail car, rubber has been used to a greater extent and for more diversified purposes than has ever been attempted in a transportation vehicle

Rubber Parts for Various Purposes

The design of the rubber parts for heat and sound insulation, sealing, abrasion resistance and resilient mountings in the light-weight rail car, Autotram

F. L. HAUSHALTER, *Development Engineer, B. F. Goodrich Rubber Company*

ALMOST every material has several physical or chemical properties that make it specially suitable for a number of different purposes. Thus, because of its resiliency, wear resistance and low heat conductivity, rubber is specially suitable for a number of different uses. These can be illustrated by the application of rubber in various products, but in the light-weight rail car built by the Clark Equipment Company, rubber has been used for the greatest variety of purposes for which it has ever been applied in any one product. The design problems involved and their solutions present not only one of the most interesting steps in the development of railroad equipment, but also one of the greatest strides towards a more general application of rubber in its various forms.

On branch lines where the traffic has fallen off to such an extent that the railroads cannot operate a regular train and crew on a profitable basis, the diesel-electric and gasoline-electric rail cars of standard construction have already made a place for themselves. These cars, built according to the typical railroad car design, are relatively heavy. But the success of their operation has resulted in considerable interest manifested by railroad men in the possible use of light-weight rail cars for both main-line and branch-line traffic. Such light-weight cars can be operated at speeds of 80 to 110 mi. per hr. and have a high rate of acceleration and deceleration. With short distances between stops, acceleration and deceleration largely determine the running time. Hence the added emphasis placed upon weight.

In the Autotram, designed and built by the Clark Equipment Company, the engineers departed almost entirely from the usual type of car construction, both in the design and the materials used. To obtain lightness, the car is constructed almost entirely of aluminum alloys. It is 60 ft. long and weighs 35,000 lb.

The body of the car is supported by two four-wheel trucks of special design. The wheels of the truck have their steel tread insulated from the hub section by rub-

ber disks or rings which are pulled up in compression against knurled metal surfaces. A cross-section of this wheel is shown in Fig. 2. The action of the rubber in this design of wheel is one of compression-shear. The amount of vertical movement in the rubber as the wheel revolves can be controlled by the hardness and thickness of the rubber disks. These rubber inserts, being in the wheel itself, afford the first cushioning between the tread and the car body. Being on the unsprung member the rubber is given the opportunity of cutting off the peaks from the shocks delivered to the axle by impact of the wheel tread with the rail joint or cross-over. Insulating the wheels in this manner also changes the pitch of the sound which is set up by the web section of a conventional car wheel. This can be readily demonstrated by striking the side of the wheel with a hammer. Reducing the pitch of these tones greatly reduces objectionable air-borne noises carried into the car body.

Inasmuch as the rubber in such a wheel would not stand up under the temperatures generated by the conventional tread-shoe brakes, it was found necessary to brake the car by the use of separate brake drums on the axles. The brakes are of the automotive type, internal expanding, and operated by air pressure. The conventional street car would greatly relieve the noise in our thoroughfares if equipped with these rubber insulated wheels.

It is well known that the noise and vibration producing element of any railroad car is the truck. On the Clark rail car the body is insulated from the trucks by specially designed center plates, as shown in Fig. 3. This center plate is perforated with holes in order to allow flow of the rubber under vertical deflection, for

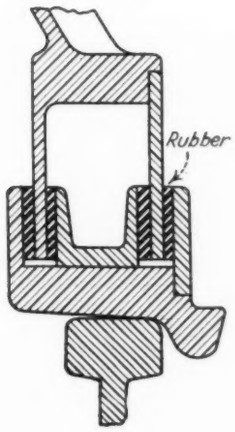
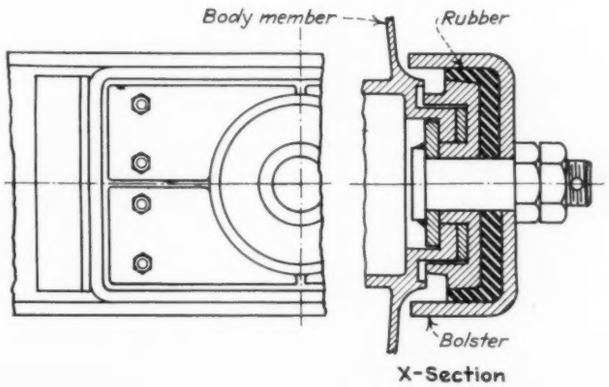


Fig. 2 — Rubber disks, compressed against knurled metal surfaces provide a cushion between wheel tread and axle. The action of the rubber is one of compression-shear and the vertical movement in the rubber as the wheel revolves depends on the hardness and thickness of the rubber disks

Fig. 3—Design of the rubber insulated center plates which cushion both vertical and side movement. To allow flow of the rubber under vertical deflection, the center plate is perforated



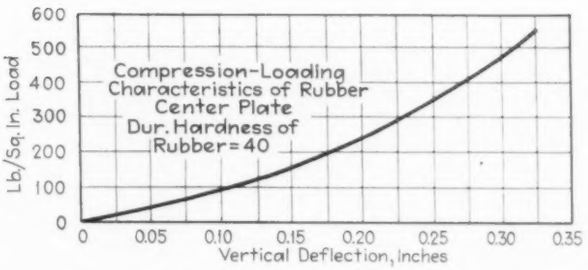


Fig. 4—Loading characteristics of the perforated live rubber used in the center plate

unless the rubber has some place to flow it is ineffective as a shock absorber. The side flanges of the rubber center plate restrain side movement of the body relative to the truck and cushion this movement considerably. But these are not the only advantages of the rubber-cushioned center plate. With standard forged or cast-steel center plates, one of the most difficult problems is lubrication. Because of the difficulty of applying it, little or no lubricant is supplied to the center plate, and the resultant wear and grinding is tolerated. The live rubber between the body and truck center plate eliminates this condition.

Although these trucks are equipped with leaf springs, it is well known that a considerable blow has to be delivered to a leaf spring before the inter-leaf friction is overcome and the spring goes into action. On hitting a rail joint or a cross-over, the period of impact is extremely short. For a cross-over the period at 60 mi. per hr. is about 1/200 sec. Live rubber placed in the center plate greatly reduces the magnitude of the shock delivered by the truck to the body before the steel springs can act. The curve in Fig. 4 shows the loading characteristic of a center plate of perforated live rubber. Under the normal weight of the car, the rubber deflects an amount equal to about 100 lb. per sq.in. compressive load. When the truck hits a badly worn cross-over, where the wheel may drop $\frac{3}{8}$ in., the unit load on the rubber is increased and the rubber deflects proportionately to cushion the blow delivered to the body.

One constant cause of noise on conventional trucks has been the side bearings which restrain body movement when a car rounds a curve or starts to sway. Unless the conventional side bearing is kept lubricated, the rubbing action with the body sill sets up an undesirable squeak. On the Clark car, tough, abrasion-resistant rubber is bonded to a metal plate, which plate is fastened to the truck structure. When the car is on a level track there is a slight clearance between the body sill and the surface of the rubber side bearings. As the car moves down the track and the body starts to sway because of irregularities in the track and rail joints, the rubber side bearings quietly steady the body of the car by a rubbing contact with the sill. The friction between the rubber and the sill dampens sharp changes in lateral body movements. The rubber also deflects a small amount and gives a relative floating action between the truck and the car body, besides having the advantage over a hardened steel side bearing of being noiseless.

Particular care was taken in the design of this car to insulate the body against vibration and noise. The aluminum panels are insulated from the aluminum structural uprights by rubber insulating strips, the pur-

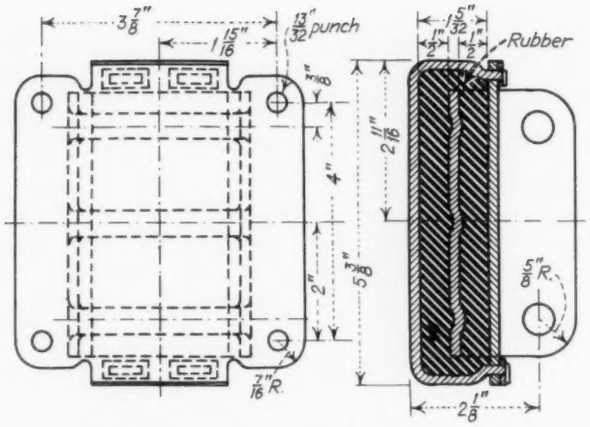


Fig. 5—Four cage-type shear rubber mountings support the motor, the central member of the mounting being attached to the motor arm

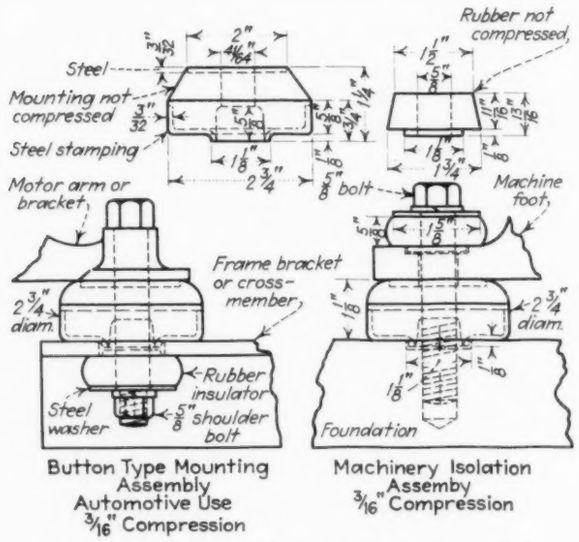


Fig. 6—Button-type rubber mountings are used for the air compressor and auxiliary electric generator

pose being to eliminate metallic connection between the outside and inside walls of the car, thus interrupting the sound and heat conducting circuits.

Because the passenger compartment of the car is air-conditioned, the windows have been permanently sealed. Double glass windows are set in U-shaped strips of black, non-blooming rubber. This simplified window construction gives a permanent seal against noise and heat by maintaining a dead air space between the two window panes. The rubber also seals against the entrance of dirt. This method of window construction suggests the type that will be used in factories, office buildings and homes designed for air-conditioning. The single window pane of glass is a poor heat insulator and the ordinary window sash is anything but dirt tight.

The floor of the car is made up of sheet aluminum, and to insulate this against heat and noise, sponge rubber $\frac{3}{8}$-in. thick is laid on top of the floor, then a carpet is placed on top of the sponge rubber. This sponge rubber, blown from highly plasticated pure gum, together with the carpet affords a quiet covering. One of the chief reasons for using this kind of insulation is that it has a highly cushioning effect on one's tread without the disadvantage of being pounded down by constant use. Such a sponge rubber cushion practically doubles the life of the carpet, and for this reason alone should find a place in home and office. The porosity of the sponge rubber makes it not only a good heat insulator but makes the rubber effective in absorbing sound.

With reference to the heat conductivity of the type of cellular rubber used, it is only 0.000089 gram calories per centimeter cube per deg. C. per sec. as compared to 0.000180 for cork and 0.350 for aluminum.

Sheet aluminum is used for the steps of the cars, a highly abrasive-resisting, non-skid rubber tread being mounted on each step. The rubber tread is bonded to a thin metal plate, and this plate is fastened to the aluminum step. When the rubber has been worn out the tread can be renewed readily. The rubber, only $\frac{5}{32}$-in. thick, is of the same quality as found on the best grade of pneumatic tires. It is entirely feasible that this type of design of renewable treads can be applied on the steps of all passenger conveyances and for stair treads for other purposes.

One novel feature of the design of this car that emphasizes the particular care that was taken in reducing air resistance to a minimum is the manner of mounting the car steps. When the car starts the steps are pulled up, thereby greatly reducing the air resistance.

The Clark Autotram is propelled by a sixteen-cylinder V-type gasoline engine which drives the front truck through a bevel gear hook-up and a special design of centrifugal clutch. The motor is insulated against vibration by four cage-type shear rubber mountings shown in Fig. 5. The rubber is bonded to brass-plated metal

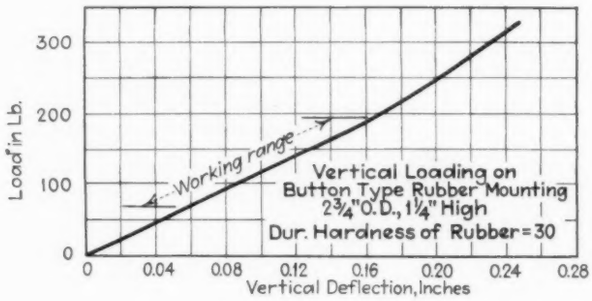


Fig. 7—Loading characteristics of the rubber as used for the button-type mountings

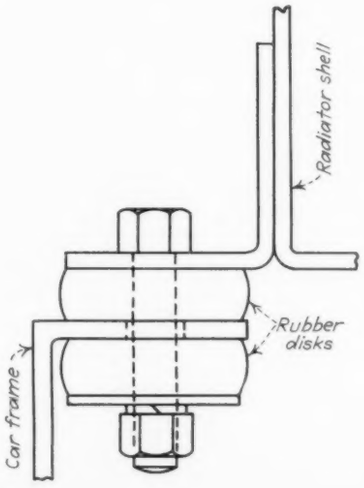


Fig. 8—The radiator is mounted on soft thick rubber disks that permit movement of radiator relative to car body

and the design is of a safe construction for heavy motors. The central metal member of the mounting is attached to the motor arm and is insulated by shear rubber (not in compression) from the other two brackets which are attached to the car frame. The loading characteristic of this mounting is a straight line similar to that of a helical spring. It is the type of rubber mounting which is gradually finding application in the field of machinery isolation. One large machine weighing 20,000 lb. has already been successfully isolated against vibration by the use of large shear-type rubber mountings.

The air compressor and auxiliary electric generator are mounted on button-type rubber mountings as shown in Fig. 6. The button-type mounting has a straight line loading characteristic similar to a shear type mounting, as shown in Fig. 7. The working range of the loading on these mountings, when properly installed, is on the straight line portion of the curve. These mountings effectively dampen vibrations set up by a unit such as an air compressor and yet afford a stable alignment because of the bolted-through method of attachment. Electric motors and motor-generator sets may be isolated by this design of rubber mountings.

The sixteen-cylinder engine is cooled by a large radiator at the front end of the car. This radiator is mounted in soft rubber as shown in Fig. 8. The rubber disks are soft and thick enough to permit sufficient movement of the radiator relative to the car body so that fatigue of the soldered joints of the radiator is greatly reduced.

Railroad men are giving more and more thought to the application of rubber to railroad cars, and the big question has been where to use it most effectively. The applications of rubber on the Clark rail car, as described here, have demonstrated in tests during the past eight months, the effectiveness of rubber as a noise, sound, and heat insulator.

Stresses and Deflections in a Built-up Beam

C. O. SANDSTROM
Thermal Engineering Company

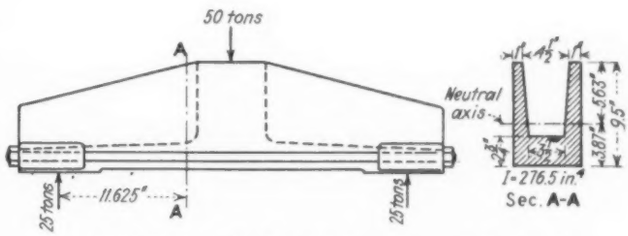
In the July *Product Engineering*, F. H. Frey submitted the design of an axle-testing fixture, shown in the diagram. The question related to the tension required in the tie-rods to deflect the fixture upward $\frac{1}{16}$ in., and also the pressure required to remove the deflection. With the nuts tightened up, the casting acts as a column, with round ends and no end restraint. With only bending moments imposed on the casting by the tie-rods the stress would be constant throughout its length, calling for a uniform cross-section. But the external load condition approximates that of a simple beam uniformly loaded and with additional compressive forces applied to its ends, producing the maximum stress and deflection at mid-span.

The critical section of the casting is at *A*. The direct compression reduces the tensile stress in the upper fibers and increases the compressive stress in the lower fibers, so that the effect is to raise the neutral axis. Assuming that the casting has the section *A-A*, the deflection can be approximated by the formula for a simple beam with a uniformly distributed load. To produce the $\frac{1}{16}$-in. deflection will require a load of 342,000 lb. However, the extreme fiber stress in tension would be considerably more than the ultimate strength of cast iron. To produce $\frac{1}{16}$ in. upward deflection the total stress in the tie-rods is 603,000 lb., giving a unit stress of 718,000 lb. per sq.in. This stress is many times the ultimate strength of the material, so it must be concluded that the rods are inadequate for the purpose, even if it were possible to deflect the cast iron to the extent stated.

If the two $\frac{7}{8}$-in. rods are of 40-carbon steel with an elastic limit of 50,000 lb. per sq.in., the force applied by these bolts should not exceed 21,000 lb. The algebraic sum of the stresses in the extreme tension fibers of the casting is then —558 lb. per sq.in., and in the compression fibers is 1,427 lb. per sq.in. The direct compression does not directly contribute to the upward deflection of the fixture, so only the bending moment determines the deflection. The probable deflection of the fixture under the 21,000-lb. force in the rods is 0.0022 in. Any attempt to remove the deflection of the fixture would increase the stress in the rods already stressed to the maximum.

The form of the test fixture is well adapted to resisting the bending stresses, because the larger area is subjected to tensile stress under which cast iron is comparatively weak, but the use of steel rods is of doubtful economy. These steel rods can carry only twice the stress of cast iron of equal area and equal distance from the neutral axis.

If the ram of the press under 50 tons pressure covers the central block of the fixture and assuming that there is no initial stress in the bolts the load deflection is 0.00592 in. The extreme fiber stress in compression is 11,460 lb. per sq.in., and the tensile fiber stress is



Steel tie rods in this test fixture are of little value in increasing the strength of the casting

7,370 lb. per sq.in. This compressive stress is rather low for cast iron, but the tensile stress is rather high. An improvement in the design would be a little more material in the tension area, or larger bolts, or both. The desired deflection, however, would be out of the question, as the corresponding stress would far exceed the ultimate strength of the material.

PRODUCT DEVELOPMENTS

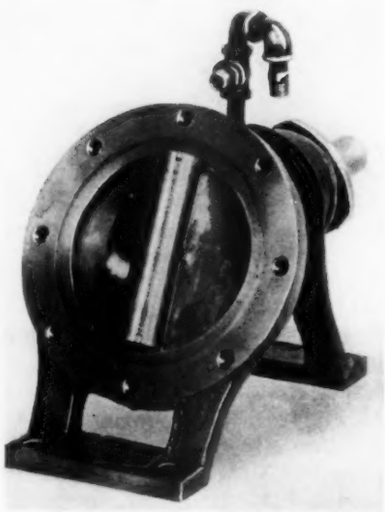
Wobble-plate pump with many adaptations—Streamlining distinguishes bus body—Mechanical ventilation permits subway car silencing—New 14-passenger Douglas airliners luxurious —Tower truck for servicing pole lights—Lead alloy for pipes

▼ ▼ ▼

▸ Pump Using Wobble-Plate Principle

An adaptation of the familiar wobble-plate pump, the new positive-displacement rotary pump, here illustrated, is of interest because of the number of possible adaptations. The unit is basically a spherical segment carried in a chamber having a spherical surface. The pump chamber is closed by an abutment with a plane face free to rotate in the inclined plane of the flange. When the segment is rotated by the horizontal shaft pinned to the cylindrical extension, the axis of the cylindrical extension is constrained to rotation in the plane of the flange.

The segment, when rotated by the shaft, follows a wobbling path which causes it to approach and recede from the closing plate. Rotating with the segment, the closing plate is provided with ports that coincide with stationary ports properly located in a cylinder head not shown. Thus suction on one face is caused simultaneously with compression on the other face of the segment or piston as the drive shaft is rotated.



In this unusual rotary pump displacement is obtained by a wobble-plate action

Tests reported from several university laboratories state that the design is sound in principle and possesses features, such as light weight, small bulk, almost complete absence of reciprocating motion, positive displacement with small clearances, and simplification of the valve mechanism.

▸ Streamlined Bus

This streamlined 17-passenger FitzJohn parlor coach, mounted on a Studebaker chassis, is a bus in capacity and stamina, but an automobile in appearance and performance. Parcels, express and heavy luggage are carried in a built-in compartment at the rear that can be serviced by one man without a ladder. Luggage in this compartment is kept out of the weather and the weight of the luggage is kept close to the ground. Inside racks above the seats are also provided for hand luggage. Roof and cowl scoops are installed for ventilation; either hot water or hot-air heating can be used. Seats regularly furnished

Appearance, comfort and performance of an automobile are features of this 17-passenger FitzJohn parlor bus

with this car have a new type of lower spring foundation designed to provide exceptional comfort. Similar coaches ranging from 13 to 33-passenger capacity are made for various chassis.

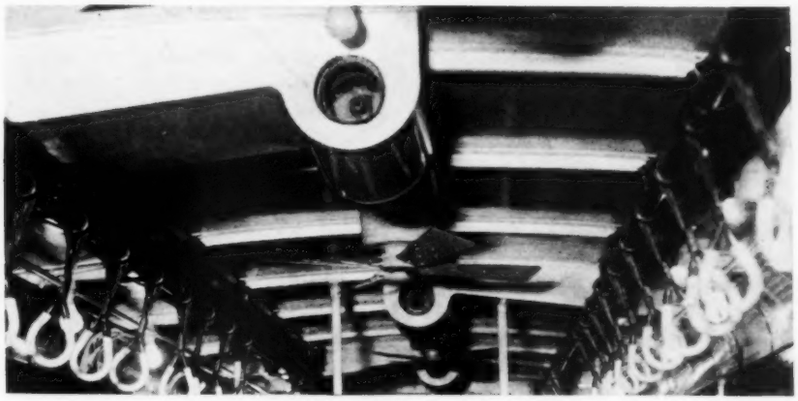
▸ Mechanically Ventilated Cars

Tests of the "noiseless" car developed by Interborough Rapid Transit (N. Y. City) engineers have been so successful that a 10-car train, sealed and mechanically ventilated, is being equipped. The cost for the ventilation per car is about $300, which would be much reduced if a large order were placed. The illustrations indicate the method of ventilating the test car, which is sealed against outside noise, so far as possible, this constituting the "noiseless" feature.

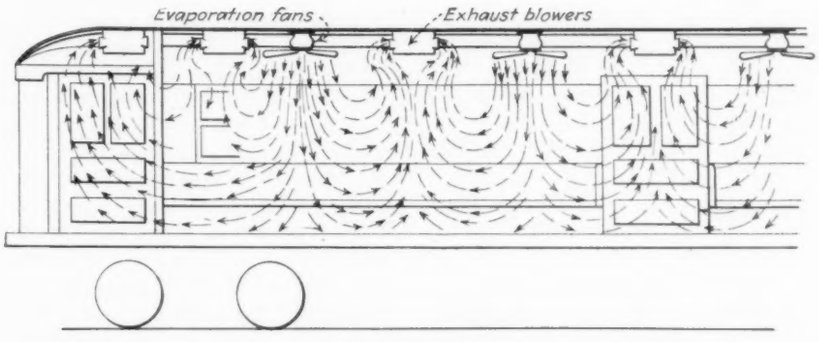
The seven Sturtevant duplex exhaust blower units, five in the car and one in each vestibule, are driven by ¼-hp. motors, and rate 714 cu.ft. of air at 1,400 r.p.m. Baffle plates shut out the entrance of sound. All doors and windows are fitted with weather strips and the ventilators are permanently closed. If the power goes off six of the upper windows are arranged to drop automatically, but only after a 3-min. interval controlled by a time delay relay.

In addition to reducing the noise level from 81 to 61.5 decibels (a Pullman car rates at 60 decibels), the installation cuts down dirt and dust and reduces carbon dioxide content about one-half.

The first of fleet of Douglas "Airliners" for Transcontinental and Western Air, Inc., embodies unusual provisions for passenger comfort as well as higher speed and greater safety



Fresh air is supplied to all parts of the interior by ceiling fans which exhaust the air from the car, fresh air entering through the crevices around doors and windows

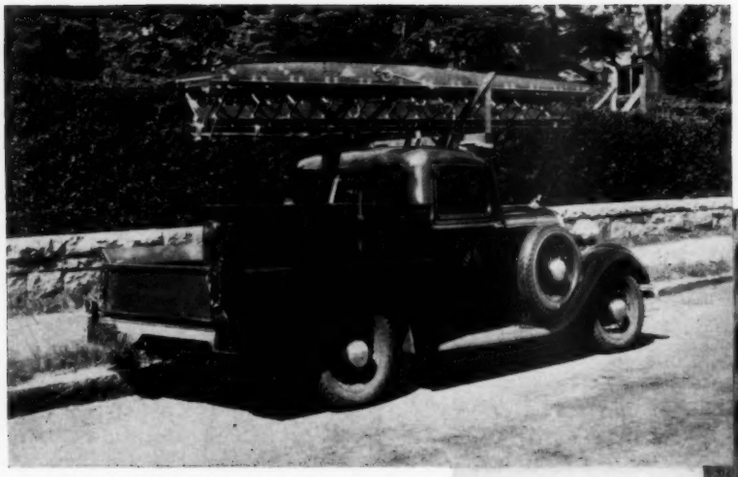


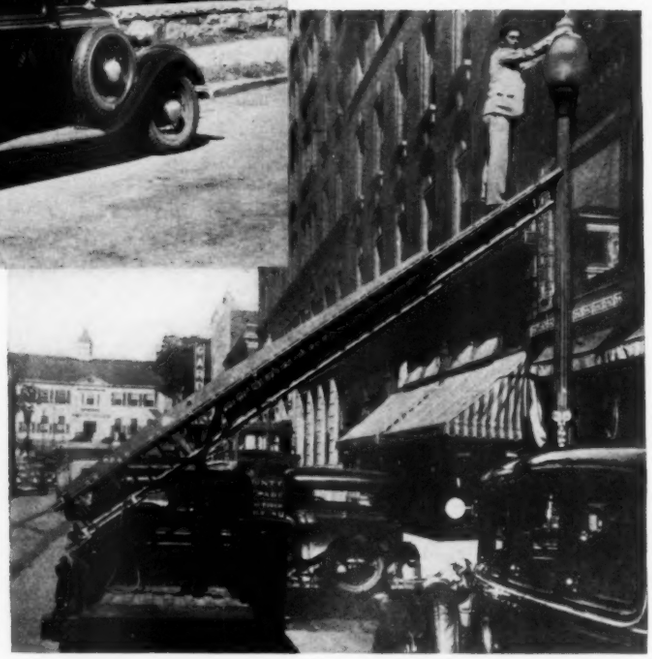
▸ New Douglas Airliners

A new standard of airplane passenger comfort is realized in the new Douglas "Airliner" recently accepted by Transcontinental and Western Air, Inc. In this low wing, all-metal cantilever monoplane, equipped with two motors, the passenger cabin is 6 ft. 3 in. high and 5 ft. 6 in. wide. Fourteen seats in two rows are spaced 40 in., back to back. Seats are mounted in rubber, having adjustable seat and back, and are reversible, allowing passengers to face either direction.

Excellent vision is said to be provided for all passengers, since the cabin passes over the top of the wing rather than through it. Individual reading lights, ash trays, literature pockets, and a mesh hat rack are installed for each passenger. Soundproofing developed by the Sperry Gyroscope Company is said to give a noise level of 70 decibels, comparing favorably with Pullman car ratings. In connection with the soundproofing a thermostatically controlled heating and ventilating system supplies air through ducts from the nose of the fuselage.

Also included for the convenience of passengers is a buffet complete with an ice box and facilities for





For servicing street lights this tower truck is both low in cost and safe. By means of the swivel hinge a street lamp over a parked car can be serviced

serving meals aloft and a completely equipped lavatory at the rear of the cabin.

The entire structure is constructed of the new 24S alloy developed by the Aluminum Company of America. The wing of cellular construction is 85-ft. in span with outer wing panels demountable. A retractable landing gear operated by a pump in the control cabin is counterbalanced and can be lowered in 20 seconds. Landing gear retracts forward with the axles resting in sockets in the main bulkhead so that the plane in an emergency can be safely landed with the gear retracted.

Engine mount and all auxiliaries forward of the firewall are quickly detachable and interchangeable right and left. Removal is facilitated by grouping connections at the firewall and by the use of detachable plugs for electrical connections. Rubber bushings for the engine mounting reduces the transmission of vibration to the structure. Elastic Stop Nuts are extensively used for holding removable cowling and fairing, gasoline tank covers and inspection plates, approximately 30,000 being used in each plane. Many parts ordinarily riveted are bolted together for easy replacement.

Fully loaded with 14 passengers, crew of two, baggage, cargo and fuel for 900-mile range, the gross weight is 17,500 lb. A cruising speed of 185 m.p.h. with a maximum speed of 205 m.p.h. is obtained under full load, with a service ceiling of 22,000 ft. Landing speed is 59 m.p.h., and one motor under full load will take the plane to an altitude of 11,000 ft.

▸ Tower Truck

In servicing present-day street lights the service man must get up to the lamp instead of lowering the lamp to the ground. A tower truck for this service must be light weight, so that only one operator is needed. In order to pass under bridges and trolley wires, the tower must be collapsible, but must be safe for the operator. In addition, electrical insulation to ground other than the tires must be provided.

To meet these requirements, the tower truck shown in the accompanying illustration was developed. The tower is mounted on a standard Ford or other inexpensive commercial truck. The extension ladder is of triangular section arranged so that the upper section telescopes inside the lower part. A hinge mechanism at the rear of the lower ladder is mounted on a swiveling pedestal secured to the frame of the car. This arrangement permits rotation of the ladder in both vertical and horizontal directions.

Two locking bolts at the base of the lower ladder section engage with holes through the floor of the box. For each of the two bolts an electric switch is arranged so that it is closed only when the bolt is in the locked position. These two switches connected in series operate an indicating light in view of the operator.

Being swiveled in two planes, the ladder can be elevated either parallel with the truck or at an angle. With this arrangement it is possible to service a street light even when a car is parked at the curb.

▸ Lead Alloy Pipes

A new alloyed lead water pipe was recently announced in England by the British Non-Ferrous Metals Research Association. It is composed of 98.25 per cent lead, 1.5 per cent tin, 0.25 per cent cadmium, or 99.25 per cent lead, 0.25 per cent cadmium, 0.50 per cent antimony. The pipes made from these alloys are one-third lighter than all-lead pipes of similar capacity, develop a tensile strength 60 per cent greater and a resistance to vibration 300 per cent, or more, greater than ordinary lead pipe.

Laminated Metals

Laminated sheets, tubes, bars or wire with a thin veneer of solid metal make it possible to use precious metals economically

GEORGE A. RUEHMLING
Manager of Engineering, General Plate Company

IN LAMINATED metals the bar, plate, tube or wire is made up of two or more layers of different metals fused together to form a solid mass. There are several considerations that may dictate the use of such veneered or laminated metal. Cost may prohibit the use of a certain metal in spite of the fact that the material in question is the only one whose properties fully meet the service requirements. Gold is corrosion-resistant to everything except aqua regia, but its cost, especially since the value of fine gold has jumped from a stable $20.67 per troy ounce to a $30 fluctuating level, usually prohibits the use of solid sheets, tubes, bars or wire of this metal.

USES OF LAMINATED METALS

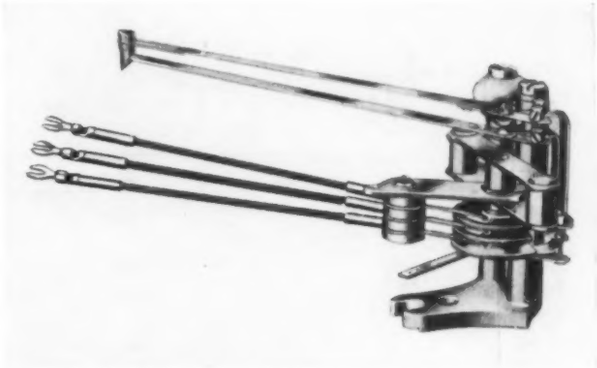
Contacts	Corrosion Resistance	Thermostatic Metal
Relays	Diaphragms	Household appliances
Motor starters	Gas appliances	Thermostats
Circuit breakers	Meters	Circuit breakers
Household appliances	Chlorinating apparatus	Time switches
Watt hour meters	Chemical equipment	Scales
Voltage regulators	Safety valves	Thermometers
Contactors		Sign flashers
Flashers		Safety pilots
Controllers		Damper controls
Signal systems		Fire alarms
Instruments		
Thermostats		
Switches		

Sometimes it is necessary or desirable to combine the qualities of two different metals. Fine silver is acknowledged to be the best metal for general contact purposes, as the oxide which forms on silver does not affect its conductive properties to any noticeable extent, but it is also known that pure silver is very soft. By using a heavy monel backing beneath a fine silver facing it is possible to impart toughness and ruggedness to the highly conductive silver layer, even at elevated temperatures, such as encountered in electric flatirons. Similarly, a thin gold veneer on a phosphor bronze spring wire gives a spring that is completely resistant to all conditions of corrosion commonly met.

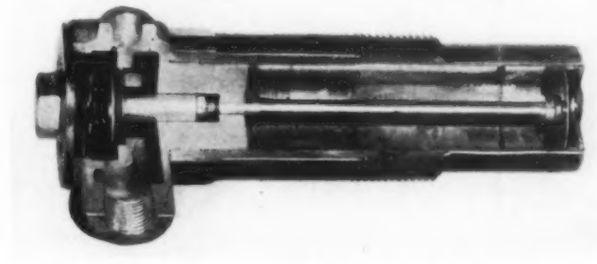
An electro-deposited layer of metals, being composed of minute deposited particles, is naturally porous and has a grain structure that offers poor wearing qualities. On the other hand, rolled plate is made by taking a heavy bar of base metal approximately 1 in. thick and welding it directly to a heavy solid sheet of precious metal. The combination bar is then rolled to the desired thickness and temper, the precious metal surface remaining solid and hard and, therefore, giving good resistance to abrasion. The proportional thicknesses of the different layers remains constant throughout the rolling operation.

A wide range of combinations of metals can be obtained and more than two layers can be fused together so that much latitude is presented in the way of selecting the metals to meet specific requirements. The united metals combine the desirable properties of both the precious metal and the base metal with the added advantage of lower cost.

Laminated metal is available not only in strips but also in the form of wire, bars and tubing. Laminated tubing is made by cupping and drawing a circle of laminated sheet into the form of a tube. By successive



Laminated platinum-iridium is used for the tip of the contact brushes and other parts of the Rotax recording instrument made by the Foxboro Company



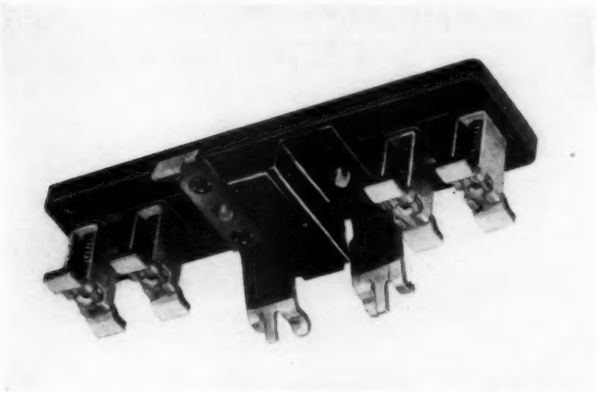
The helical spring in this Spencer thermostat is gold plate on bronze wire, the thermostatic disk is brass-Invar

Properties of Corrosion-Resisting Alloys

| Metal or Alloy | Composition | Condition | Ultimate Strength | Per Cent Elong. in 2-in. | Specific Gravity | Scaling Temp. deg. F. | Annealing Temp. deg. F. | Corrosion-Resistant Properties (See Note) | | |

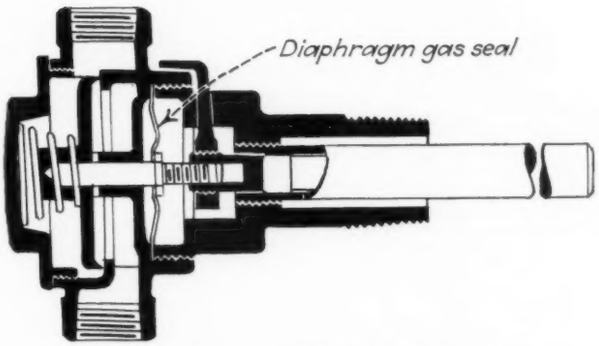
drawing operations the tube is reduced in size to the desired diameter and wall thickness. Wire is produced in a similar manner by inserting and welding a core of base metal in a tube of precious metal of a relatively large diameter and then reducing the solid combination to the desired diameter of the wire.

Another form of laminated metal is strip inlaid with precious metal stripes. In manufacturing this striped material, longitudinal grooves are milled in one face of a large bar of base metal. Precious metal bars are snugly fitted and welded securely into the grooves. The composite bar is then rolled into a strip of the desired cross-section. Such strips will withstand severe form-



Courtesy Cutler-Hammer, Inc.

For electrical contacts, laminated silver is superior to electroplated silver, and is usually cheaper



In this Hotstream thermostat the diaphram is of acid-resisting gold plate to prevent corrosion

ing operations without any separation taking place between the stripes and base material.

Striped material is being used in a great number of electrical contact applications. Silver has several properties that make it particularly suitable for electrical contacts and the striped material effects economies by having the silver in only those portions where it is useful. Furthermore, laminated metal contact units offer improved performance and service by reason of the fact that the one-piece unit conducts heat away from the points of contact more rapidly than riveted contacts, and there is also no possibility of experiencing troubles caused by the loosening of parts as in multiple-piece riveted assemblies.

Another interesting electrical application of laminated metal is in the ordinary electric lamp bulb. Solid copper wire cannot be used for the lead-in wires that are sealed in the glass near the base of the bulb, because of the difference in the coefficient of expansion between glass and copper. Platinum was used at one time for this purpose. The coefficient of expansion of some nickel steels is close to that of glass, but the glass will not adhere to the steel. By using a wire having a core of nickel steel with a thin outside sheath of copper, the expansion factor is taken care of and at the same time a tight seal is obtained betwen the glass and the metal.

In the accompanying table is given a list of the metals and alloys ordinarily used in laminated metal. Almost innumerable combinations are possible and there are many factors that enter into the selection of the particular metal to use for a given application. Generally speaking, it is not economical to laminate two base metals, such as pure nickel and brass, with the object of producing a material cheaper than either one of the two metals in solid form. Where the price differential betwen the two metals is small, the cost of laminating the metals will offset the price differential.

In the accompanying illustrations are shown several applications of laminated metal. Most of these examples illustrate the use of this material for corrosion resistance. Another big field for the use of this material is for thermostatic bimetal.

Thermostatic metal or bimetal is in the form of a solid laminated sheet composed of two metals. One metal has a high coefficient of expansion, while the other metal has a low coefficient. The two metals in the form of heavy sheets are welded together into a composite block and rolled to a thin strip. The strip, cold-rolled to the required thickness, is punched, bent, or formed to the required shape.

When thermostatic metal is heated the high-expansion material elongates more than the other metal. Since they are solidly joined, the composite strip is compelled to bend. In every design of thermostatic element, for a given change in temperature, the amount of movement, the force that can be exerted, and the allowable ranges of temperature depend on the metals used.

The problem of selecting the best combination of metals for a given bimetal application is a complicated one. The amount of movement obtained will be proportional to the difference between the coefficients of expansion of the two metals. But with every metal the coefficient of expansion changes at different temperatures. For example, Invar has a coefficient of expansion of practically zero in the temperature range from normal room temperatures up to about 300 deg. F. At higher temperatures the coefficient of expansion rapidly increases and soon approaches that of carbon steel. Increasing the nickel content of this alloy results in a low coefficient of expansion being possessed by the alloy up to a temperature of approximately 500 deg. F. Substituting cobalt for a part of the nickel in this high-nickel alloy further decreases the coefficient of expansion throughout the temperature range.

It is quite apparent that careful thought must be given to the selection of the proper metals for the different layers if best results and maximum benefits are to be derived from the application of laminated metal to specific problems.

Graphical Solutions For Helical Gear Problems

For a given helical gear design an indefinite number of helix angle and gear diameter combinations are possible. Cut and try numerical methods are eliminated by this graphical solution of the helical gear formula

J. K. OLSEN
Chief Draftsman, Stewart-Warner Corporation

HELICAL gearing is the present standardized term applying to gear combinations in which the gear teeth form true helices. To maintain true helical form in the teeth of a gear combination requires that the two shafts lie in parallel planes, although the shaft axes may or may not be parallel. In many applications the shafts are at 90 deg. to each other, the helix angle being 45 deg., and the pitch line speed the same for both gears. Helix angles less than 20 deg. are seldom used.

With the helical gears on non-parallel shafts, two methods of changing speed ratio are possible. As with spur gears, the diameters can be changed, using the same tooth helix angle. When this method is used, the ratio of pitch line speeds remains the same. The other method is to change the angle of the tooth helix, thus changing the number of teeth without changing the diameter. In this method the pitch line speeds of the two gears are changed.

For a given speed ratio, shaft angle and driving gear diameter, there are an indefinite number of combinations of helix angle and driven gear diameters, with the center distance varying with every combination of diameter. It is apparent that no direct mathematical solution for a helical gear combination is possible. Handbooks give cut-and-try methods involving numerous calculations.

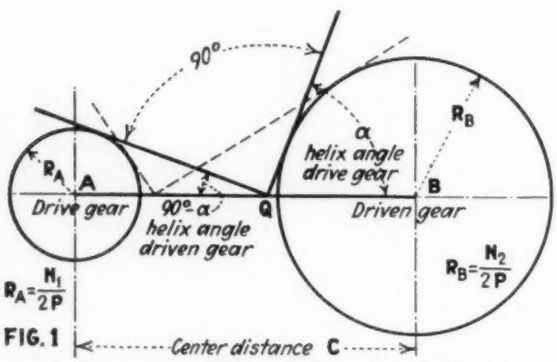
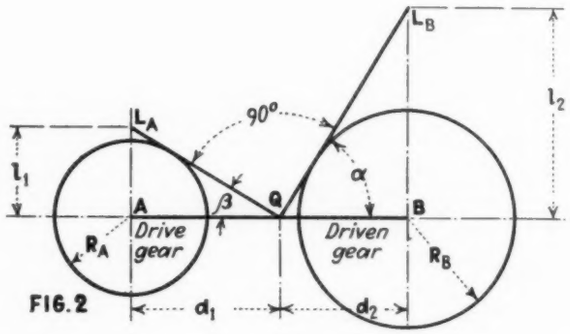


Fig. 1—By the graphical construction described in the text, the helix angle of both the drive gear and the driven gear can be readily found when the center distance, gear ratio and shaft angles are known

The formula for helical gears, in which D is the pitch diameter, N the number of teeth, P the diametral pitch, and S the helix angle, is based on the spur gear relation:

$$D = \frac{N \sec S}{P} = \frac{N}{P \cos S}$$



	Driver	Driven
Lead	$2\pi l_1$	$2\pi l_2$
Pitch Diameter	$2d_1$	$2d_2$

Check Equations:
$2d_1 = N_1/P\cos\alpha$ $\quad 2d_2 = N_2/P\cos\beta$

For the special case of shafts at 90 deg. $\beta = 90-\alpha$

Fig. 2—Mr. Olsen's graphical method also gives the lead and pitch diameter. Values obtained by the graphical method can be checked by substituting in the check equations

Helical gear design formulas based on this relation are given in the accompanying table.

For determining the helix angle the graphical solu-

tions shown here are much quicker than the usual cut-and-try methods, and when drawn to a large scale, provide a result with sufficient accuracy for most engineering applications. In Fig. 1 is given the graphical construction for the special case of shafts at right angle. The procedure is as follows:

Draw the line AB representing the center distance to as large a scale as convenient.

With the center at A draw a circle with the radius R_A equal to $N_1/2P$; at B draw a circle with radius R_B equal to $N_2/2P$ using the notations given in the table, and the same scale as used in laying off AB.

On a separate piece of transparent paper, draw two lines forming a 90-deg. angle, and place the paper over the diagram with the vertex of the angle on the line AB, and with the two sides of the angle tangent to the two circles. Find the two points where the vertex of the angle meets the center line when these conditions are satisfied.

Draw tangents to circle A from the two points found on AB and select as the point Q the intersection of the tangent making the smaller angle with the center line. Draw a tangent from point Q to circle B.

The angle between AB and the tangent to circle B is the drive gear helix angle; the driven gear helix angle is between AB and the tangent to circle A.

Basic Formulas for Design of Helical Gears

$$C = 1/2\ (D_A + D_B) \quad (1)$$

$$D_A = N_1/P \cos \alpha;\ D_B = N_2/P \cos \beta \quad (2)$$

$$C = N_1/2P \cos \alpha + N_2/2P \cos \beta \quad (3)$$

D_A = pitch diameter for drive gear
D_B = pitch diameter for driven gear
C = center distance
P = diametral pitch
N_1 = number of teeth for drive gear
N_2 = number of teeth for driven gear
α = helix angle for drive gear
β = helix angle for driven gear
For shafts at an angle of 90 deg., $\beta = 90 - \alpha$

Lead for both of the gears can be found from this layout by continuing the tangents to L_A and L_B, the points of intersection with the vertical center lines through A and B as shown in Fig. 2. The drive gear lead is the distance l_1 multiplied by 2π, and the driven gear lead is l_2 multiplied by 2π. The pitch diameter from this construction is $2d_1$ for the drive gear, and $2d_2$ for the driven gear.

The helix angle can be measured with a protractor or determined from the relation:

$$\cot \alpha = l_1/d_1$$

To check the accuracy of the graphical work the helix angle and the pitch diameter can be substituted in Equation (2) given in the table. Of course, when scaling, the same scale must be used throughout.

When the circles are too close together to place the angle so that the vertex falls on AB, no combination is possible, and one or more of the following changes must be made:

1. Increase the center distance.
2. Increase the diametral pitch.
3. Decrease the number of teeth.

COMING ARTICLES

Hydraulic versus Mechanical Feeds—*A. L. DeLeeuw, well-known consulting engineer on the design of machine tools.*

Plastic Materials—*H. M. Richardson, applications engineer of the plastic department of the General Electric Company.*

Design of Thermostatic Bimetal Elements—*H. D. Matthews, chief engineer of the W. M. Chace Valve Company.*

November—PRODUCT DEVELOPMENT SURVEY

▼

In many problems the center distance cannot be changed. Although the diametral pitch usually can be varied over a wide range, a pitch too fine results in a weak tooth, while a pitch too coarse tends to increase the noise. The tooth combination can be changed easily with small ratios such as 1:1, 1:2, or 1:3, but changes are more difficult when ratios such as 12:13, or 13:14 are required.

When a problem includes the selection of diametral pitch and number of gear teeth with the ratio given, the selection of the pitch is a matter of experience, using a pitch that has proved satisfactory in similar cases. In finding the best tooth combinations, the graphical method given in Fig. 3 is of assistance. Draw the two circles A and B so that R_A/R_B is the required ratio. Select a scale that will give a small driven gear helix angle. Using the formulas in Fig. 3, vary the pitch P, and the radii R_A and R_B until N_1 and N_2 become whole numbers. The direction of helix can be either right- or left-hand.

When the shaft angle is not 90 deg., the method of calculating the helix angle differs with various layouts. Such factors as the direction of rotation and direction of spiral and end thrust influence the choice of the proper method. On the drawings, Figs. 4 to 7, the arrows indicate the direction of rotation, the direction of the helix is indicated as left-hand or right-hand, by the letters L.H. and R.H., and the direction of thrust is shown by a thrust bearing. In the figures,

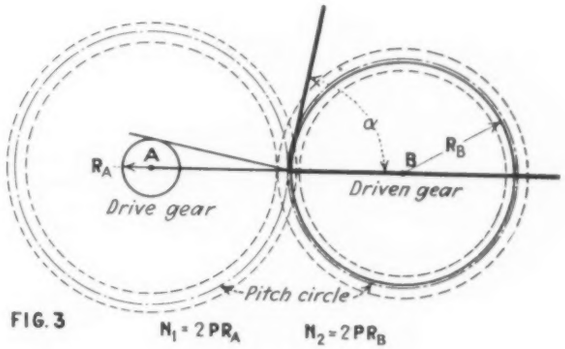


Fig. 3—The same graphical construction provides a method of determining the best tooth and diametral pitch combinations for a given ratio

α is the drive gear helix angle, β is the driven gear helix angle, and γ is the shaft angle.

Formulas governing the relation between the number of teeth, pitch, pitch diameters, helix angles and center distances for helical gears are the same as the relations given in the table. Calculation diagrams for the solution of Equation (3) differ from the solution previously given in that the angle between the lines drawn on transparent paper is not 90 deg. but is the shaft angle, γ. There are four calculation diagrams all of the same form except that the position of the tangents is adapted to various combinations of conditions. Referring to Fig. 4, the distance AB and the two circles are constructed as before. On a separate piece of transparent paper the two lines are drawn, making an angle γ, the small angle between the gear shafts. The point Q and the tangents to the circles A and B are also drawn as shown previously. The drive gear helix angle is the angle between the line AB and the perpendicular to the tangent at A; the driven gear helix angle is between the center line and the perpendicular to the tangent at B. However, several different solutions are possible by placing the lines over the diagram in different ways. The correct solution is governed by the direction of the helix and the direction of rotation as given in Figs. 4 to 7 for the four possible conditions. If two points are found which satisfy the problem, the one that gives the smaller driven gear helix angle is selected.

The angles α and β can be determined with greater accuracy by measuring the angle that the tangent makes with the center line, rather than by drawing the perpendicular to the line. Referring to the table of equations, when the angles α and β are found, they are checked by substituting into the Equation (2). Center distance C, Equation (3), should be somewhat smaller than the actual center distance to give clearance between the gears. Pitch diameters are calculated from equations given in the table, but approximate values can be found by scaling the diagram, as shown in Fig. 2. The helix angle can then be determined from the pitch diameter and number of teeth, and the lead can be determined as given in the case for the shafts at right angles.

Figs. 4 to 7—The graphical construction used with shafts at 90-deg. angle is modified as shown in these illustrations for various combinations of shaft angle and direction of rotation. Direction of rotation is indicated for each gear and direction of thrust is indicated by a thrust bearing

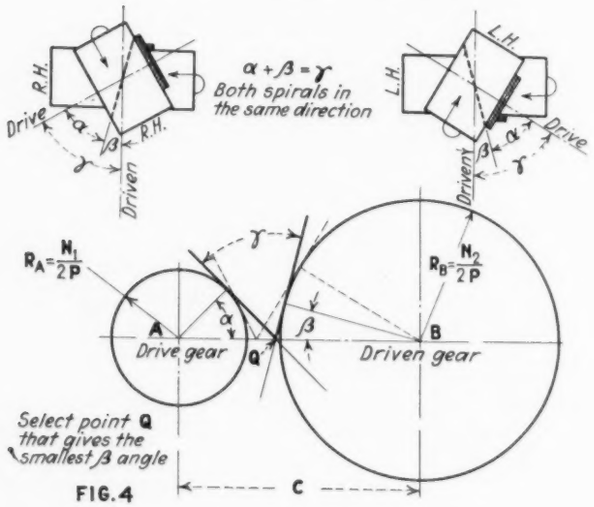


FIG. 4

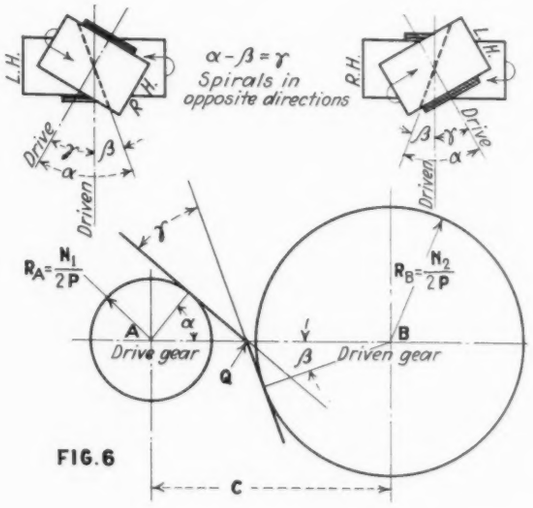


FIG. 6

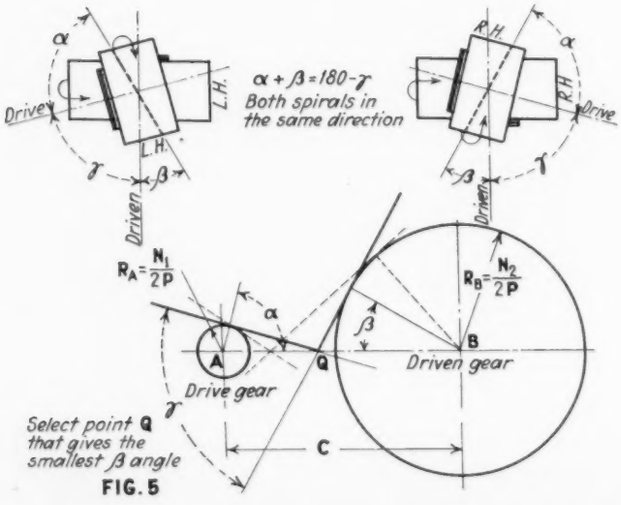


FIG. 5

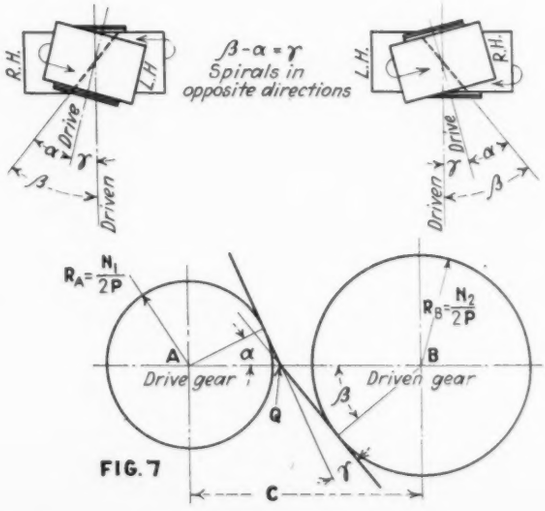


FIG. 7

Transmission Ratio and Slip of V-Belts

A report of special tests made to determine the relative importance of creep, slip and diameter effect on transmission ratios

C. A. NORMAN, *Professor of Machine Design, Ohio State University*

IT IS well known that the transmission ratio of belt drives is not a fixed or exact quantity. To begin with the pitch diameters are somewhat uncertain Even with flat belts the crowning of the pulleys and the thickness of the belt have an influence on the size of the pitch diameter which cannot be exactly given. With V-belts the depth to which the belt sinks into the groove and the location of the zone on the belt in which it bears most heavily on the side of the groove are rather uncertain. In addition there is slip. With certain kinds of leather belts a small amount of real slip may improve transmissive power and is not objectionable. With other types of belts there should perhaps be no real slip, but creep is unavoidable. With creep there is no slip where the belt runs onto the pulley; but as the belt lengthens or contracts in passing from the slack-side tension to the tight-side tension, or vice versa, the leaving part of the belt runs ahead of the pulley at the driven end and behind the pulley at the driving end, thus causing a speed loss in the drive. This speed loss is directly proportional to the elastic stretch of the belt and to the effective pull transmitted and usually amounts to about 1 per cent at rated load. The real slip and the creep together may in the case of oak-tanned leather belts be somewhat more than this The uncertainty arising from inexactness of pitch diameters is ordinarily less.

Small But Important

Small as these amounts may seem, they have a considerable practical importance. In the first place, there are drives—textiles drives, some types of paper mill drives, electric generator drives—in which, though absolute exactness is not necessary, a rather high degree of exactness of speed ratio and freedom from speed variation are essential. Then, slip is often taken by the practical man as an indicator of whether the drive is running properly or not. If the driven machine does not quite come up to expected speed, the drive is held to be no good and complaints are the result. It is essential that the designer should have some knowledge of what he can reasonably expect.

In the case of V-belts—and certian more recent types of flat belts—some harm has been done by sales organizations making exorbitant claims for their wares in regard to the absence of slip. One is told that their belt "is glued to the pulley" or "geared to the pulley"; that it has no slip whatever; that certain V-belts, fan belts particularly, never need to be tightened at all, because "they wedge themselves in the groove." Now, since creep is always present and some V-belts may slip disastrously when run very slack under maximum rated load, it is of importance that an attempt should be made to spread some correct knowledge about transmission ratios for such belts.

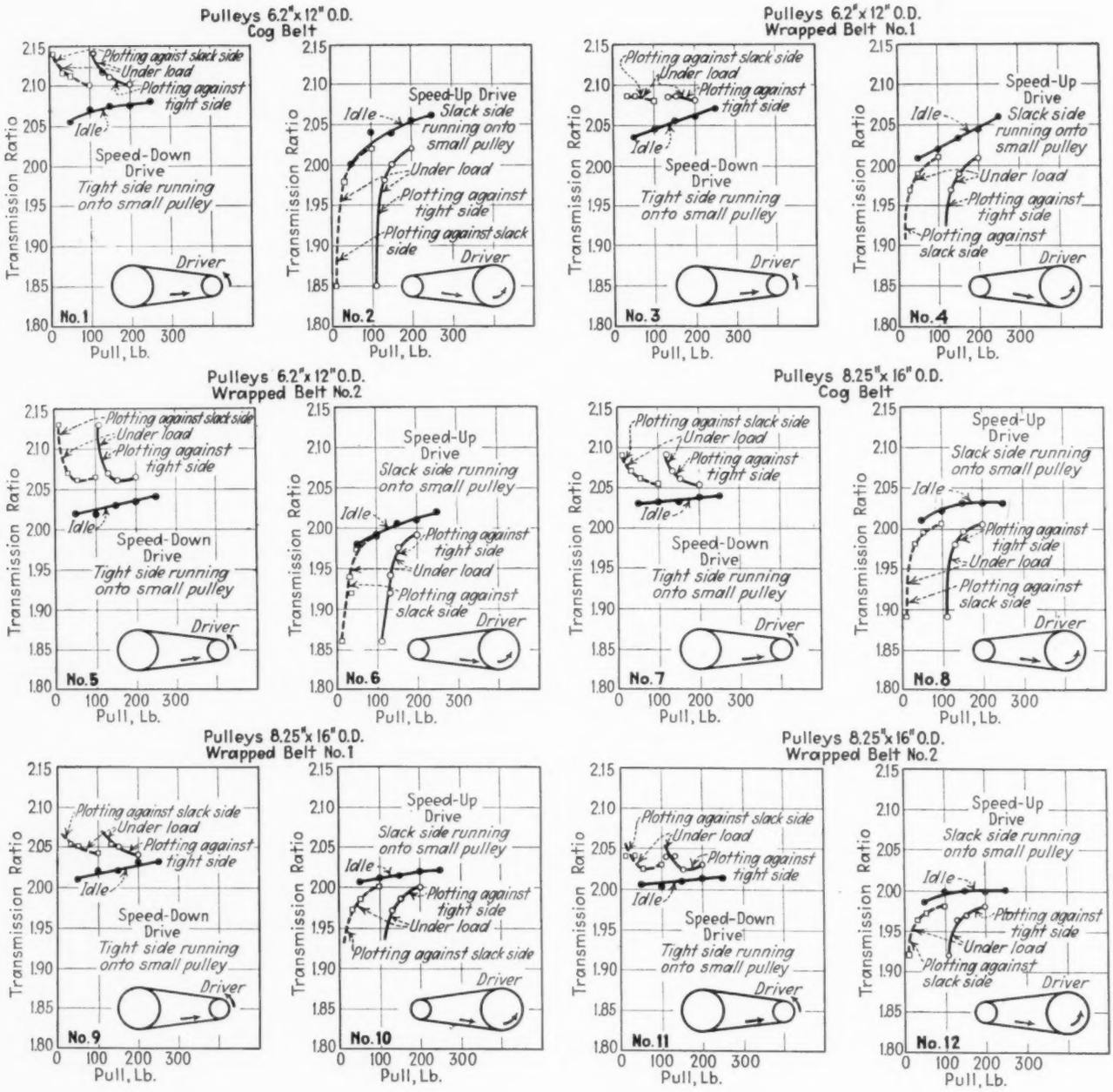
German Tests on Belt Pull

In this connection certain contentions of a prominent German engineering teacher and belt expert, Prof. K. Kutzbach of Dresden, deserve consideration. In an article reporting a very extensive research on commercial V-belts he puts exceptional emphasis on the variation of diameter with variations in belt pull, particularly with small pulleys.* In this country the practice has developed to specify V-groove pulleys by pitch diameter, or at least to give fixed corrections for deriving pitch diameter from outside diameter. If Professor Kutzbach's contentions are justified, this practice is incorrect. Professor Kutzbach emphasizes that the belt runs slack into the driven pulley, but tight onto the driving. In consequence, he says, the belt wedges itself deeper into the groove on the driving pulley than on the driven pulley. The effect is that even with pulleys of the same size the pitch diameter of the driven pulley is greater than that of the driving pulley and the speed ratio is less than unity. This effect is classed as creep or slip, but of course is nothing of the kind. Professor Kutzbach gives a tabulation estimating that this speed loss for small pulleys may amount to little less than 3 per cent.

Unfortunately no direct experimental verification of this estimate is given; and, in fact, since the pitch diameters of running drives are hardly capable of direct determination, a direct experimental verification is difficult.

The writer has therefore gone into this matter in a

*See Zeitschrift des Vereines deutscher Ingenieure, March 4, 1933, p. 240.



Graphical presentation of results of tests of three V-belts to determine effect of slip and creep on transmission ratio

somewhat indirect fashion, and the result of his investigations are herewith submitted.

The apparatus used in these investigations was the 40-hp. belt dynamometer set in the Mechanical Engineering Department of the Ohio State University. This apparatus has been described repeatedly in publications* and no description of the dynamometer set itself will be given here.

Suffice it to say that two sets of pulleys were used, one set having outside diameters of 6.2 and 12 in., respectively, another having diameters of 8.25 and 16 in., respectively; and that the speeds were determined by speed counters connected in and out by electromagnets in series so that maximum possible exactness of the speed ratios could be secured. The belts used were two wrapped V-belts manufactured by well-known concerns, and one bare cog belt, likewise of a well-known make. The wrapped belts had a maximum width of $\frac{7}{8}$ in. The cog belt rode somewhat higher in the groove and was slightly wider. The pulley groove angle was 38 deg.

These drives were run both with the small pulley as the driver, and with the large pulley as the driver, the same identical pulleys being used in both positions. To begin with the drives were idled, but under varying total pull. The higher the pull, the deeper the belts sank into the grooves. This reduced the pitch diameter on the smaller pulley by a greater fraction of the diameter itself than it did on the larger pulley, even if

*See for instance Trans. A.S.M.E., June 30, 1932, MSP-54-7, or Bulletin No. 62 of the Engineering Experiment Station of the Ohio State University at Columbus, Ohio.

the absolute amount of reduction was the same on both pulleys. In consequence the transmission ratio changed with increased tightening. (See the curves marked *idle* in Figs. 1 to 12.)

The drives were then run under an effective pull of about 100 lb., which corresponds nearly to the rated effective pull of these belts. This effective pull introduced creep. The total tension was varied so as to give a minimum tension ratio of about 2—the tension ratio at the greatest tightness which practically should be used—and a maximum tension ratio of about 11. At this tension ratio one of the belts slipped, *i.e.*, it exceeded the permissible slackness.

Factors Affecting Transmission Ratios

The speeds held were approximately 700 r.p.m. at the slow-speed end and 1,400 r.p.m. at the high-speed end. At the latter speed the torque necessary to overcome air resistance and bearing resistance of the driven armature was sufficient to affect the idling curves somewhat. That is, the idling curves for the speed-up drives show a somewhat lower transmission ratio than those for the speed-down drives.

In Figs. 1 to 12 the observed transmission ratios have been plotted against the tension in one side of the belt. For idling this is one-half of the total tension and there is one curve only. With load the question arises whether it is the tight side tension or the slack side tension, the running-on side or the running-off side that determines transmission ratio. Professor Kutzbach takes the attitude that it is the running-on side that determines it. This is the slack side on the driven pulley and the tight side on the driving pulley. The observed transmission ratios have therefore been plotted against both sides to see which gives the more reasonable result.

In judging the amount by which the transmission ratio may change by belt tightening, it seems advisable to go by the idling curves for the speed-down drives rather than by those for the speed-up drives, the latter containing undoubtedly some creep due to air and bearing resistance at the driven end, as already pointed out. It will be seen then that with 6.2-in. pulleys the transmission ratio may increase some 0.02 to 0.04 for a transmission ratio of about 2. That is, the increase is 1 to 2 per cent. Pulleys 6.2-in. in diameter are, however, entirely too small for good efficiency and long life with the belts used, and 8-in. diameters should be regarded as the absolute minimum. With 8.25-in. pulleys the change of transmission ratio within the range of tensions covered is about 0.5 per cent for the cog belt and the wrapped belt No. 2, and about 1 per cent for the wrapped belt No. 1, which evidently had a more easily distorted section.

The fact that the influence of total pull on transmission ratio becomes rather excessive on too small pulleys might perhaps be used as a quick and convenient way for settling whether or not a certain pulley size is too small for a certain belt. Determination of efficiency or life either requires complicated apparatus or else it takes a long time, while to improvise an idling rig which makes possible the measurement of total pull and transmission ratio is a much easier matter.

Plotting the transmission ratio under load on the same diagram with the idling ratios should show the creep as a reasonable difference between the two kinds of curves. In Figs. 1 to 12 the plotting has, as stated, been carried out both against the tight-side and the slack-side tension to see which plotting gives the desired result. The plots for the speed-up drives, Figs. 2, 4, 6, 8, 10 and 12, seem to show quite clearly that it is the tight-side tension which gives the more reasonable arrangement. Plotting against the slack-side tension tends to run the curve under load almost into the idling curve, showing hardly any creep at all. Plotting against the tight-side tension shows a creep of about 1 per cent or slightly more, the creep increasing as the belt is slackened up. This increase is, to begin with, no doubt due to the fact that the elastic stretch in rubberized fabrics for any given tension increase is usually found to increase as the total tension is decreased. As the belt is further slackened real slips set in and in some cases the belt could not carry the load at all at the lowest total tension used.

Since with speed-up drives the slack side runs onto the smaller pulley, it seems that it is not the running-on side that determines transmission ratio, but the tight side.

Creep Effect Predominates

The speed-down drives are interesting, because they show that at least in the majority of cases the creep predominates over the diameter effect. In the case of a speed-up drive, slackening the belt slows up the faster pulley, both because its pitch diameter is increased, and because creep or slip is increased. In other words, both effects work in the same direction on the transmission ratio.

In the case of speed-down drives, however, slackening the belt speeds up the driven pulley by diameter effect—the diameter of the smaller, driving pulley increases—but slows it down by increased creep. Slowing down the pulley means increased transmission ratio, and we find the transmission ratio in general increasing with lowered tension under load. The curves for the wrapped belt No. 2 show a somewhat irregular run in this respect, indicating that with this belt the creep does not increase as rapidly when the belt is slackened up as it does for the other belts.

Conclusions

1. The transmission ratio of V-belts is affected by the tightness of the belt at least when used over pulleys of unequal diameter. The belt sinks deeper into the groove as it is tightened, and this effects a greater percentage reduction of pitch diameter on the smaller pulley than it does on the larger.

2. This diameter effect is less serious on pulleys of reasonably large diameter than on very small pulleys, and it may perhaps be well not to use pulley diameters so small that the effect becomes pronounced.

3. On the whole, creep or slip seems to preponderate over diameter effect as the belt is slackened, but all belts do not behave exactly alike in this respect.

ACKNOWLEDGMENT—*The writer was assisted very efficiently in carrying out and plotting these tests by C. A. Hempstead, senior mechanical engineering student at the Ohio State University.*

Shifting Mechanisms

Typical linkages for the axial movement

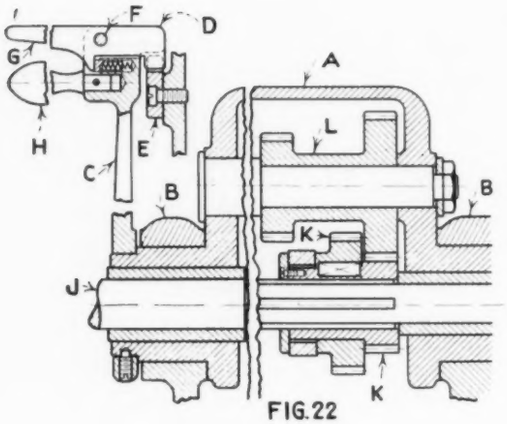


FIG. 22

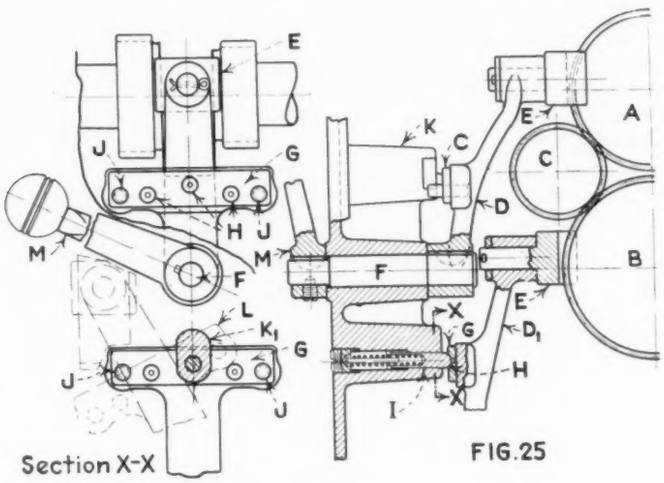


FIG. 25

ADAM FREDERICKS

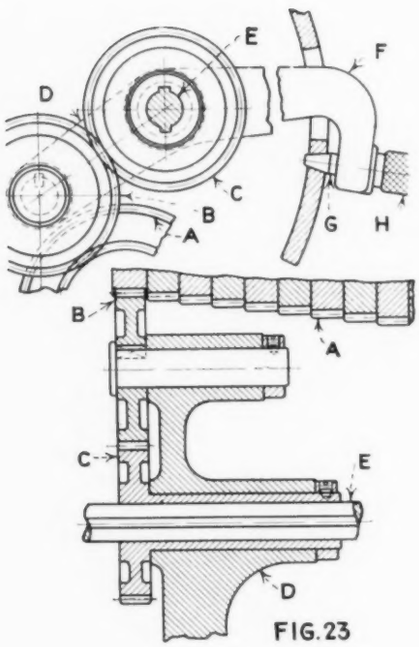


FIG. 23

Fig. 22—Gear housing *A* is free to turn in bearings *B-B*. Lever *C* rocks the whole gear case. Outside locking pawl *D* engages *E* which is screwed to the outside of the main castings. Pawl *D* is disengaged by lever *G* pivoted at *F*. Gears *K* are mounted on driving shaft *J* and engage cluster *L* which in turn mesh with sliding gear on the driven shaft.

Fig. 23—Single tumbler gear. Tumbler gear *B* engages change gears *A*. Driven gear *C* meshes with gear *B* which rotates in bracket *D* and drives the splined shaft *E*. Lever projection *F* carries locking plunger *G* and shifting handle *H*.

Fig. 24—Two-speed reversing gears shifted by hollow tube *A* which rides between the two gears of cluster *B*.

Fig. 25—Sliding members *A* and *B* are folded about gear shaft *C*. Shifting lever *D* and D_1 carry steel shifting forks *E* and pivot in the gear box cover on shaft *F*. Positioning plate *G* has countersunk spots *H* to engage spring-backed plunger *I*. Pins *J-J* have long heads to limit travel of gears in either direction by contacting with bosses *K* and K_1.

Fig. 26—Cluster *A* on shaft *B* is moved by fork *C* swiveled to lever *D*. Levers *F* and *D* are pinned to shaft *E*. Knurled handle *G* carries spring-backed locating plunger *H*. An identical unit actuates the other gear cluster. Levers *D* and D_1 are shaped to avoid interference with the bosses and to make height *J* a minimum.

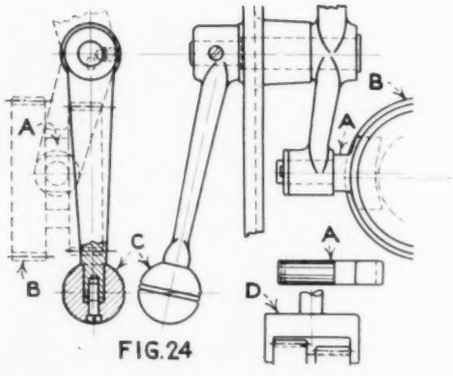


FIG. 24

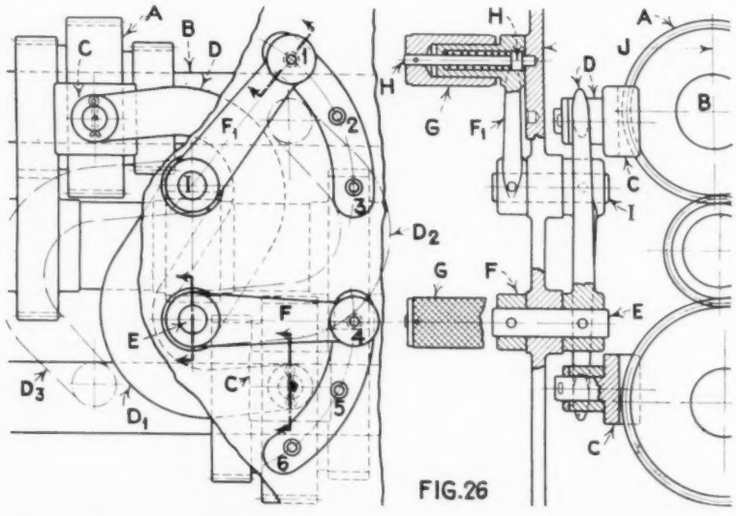


FIG. 26

for Gears and Clutches—III

of gears, clutches, wheels or other machine members

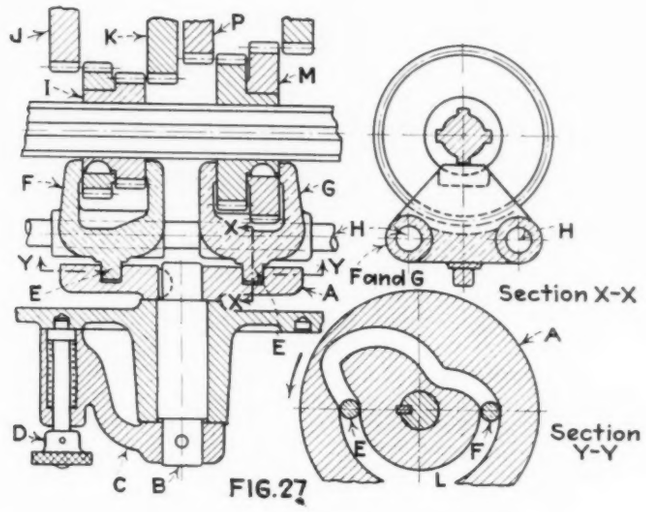


FIG. 27

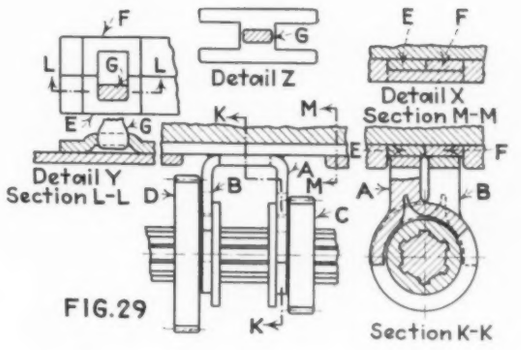


FIG. 29

Fig. 27—Cam *A* and lever *C* are mounted on shaft *B*. *E* and *D* are pin projections on gear shifting yokes *F* and *G*, sliding on rods *H* and *H*. Rotating cam *A* in direction of arrow 30 deg., 90 deg. and 60 deg., successively moves yoke *F* to cause gear cluster *I* to engage gear *J*, then gear *K*, and finally moves *I* into neutral, the position shown. During this 180 deg. movement of the cam, pin of yoke *G* rides on the 180 deg. dwell *L*. Continued rotation of cam successively causes *M* to engage gear *N* and then gear *P*, while pin *F* rides on the dwell.

Fig. 28—For light-duty, diving, sliding or plunger keys can be used. The change gears are separated by hardened and ground washers, *A*. Hardened sliding key *B* is pivoted at *C*. *D* presses *B* against bottom of keyway. Circular rack in end of shifting rod *F* engages pinion *G* fastened to hollow shaft *I* to which is attached shifting lever *J*. Rod *L* carries pinion *M* to operate another similar arrangement.

Fig. 29—In this design there are two positions of engagement for each gear. Clutch yokes *A* and *B* move gears *C* and *D* respectively, the yokes being welded to slide rods *E* and *F*. Shifting lever *G* engages slide rods as shown at view *Y*.

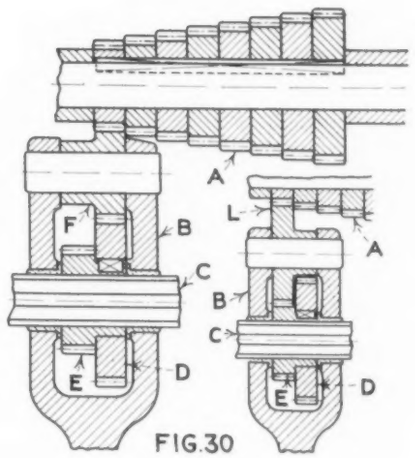


FIG. 30

Fig. 30—Longitudinal view of change gear box using two gears *L* and *D* in the tumbler mechanism. A cross-section of this design is shown in Fig. 31.

Fig. 31—Gears *D* and *E*, mounted on rocker arm *B*, are driven by splined shaft *C*. Pinion *F* on arm *B* is driven by *D* and engages any gear in the bank of change gears. Locking plunger *G* in handle *H* on lever *I* engages holes J_1. For driving through other idler gear the plunger *G* engages holes *K*.

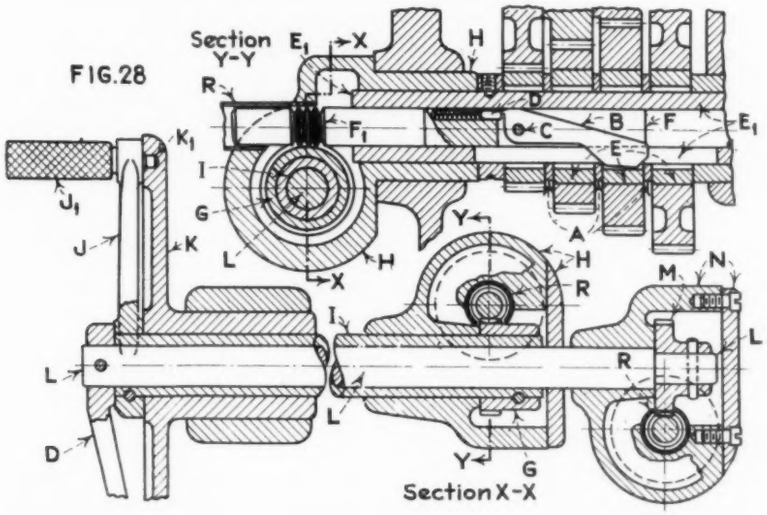


FIG. 28

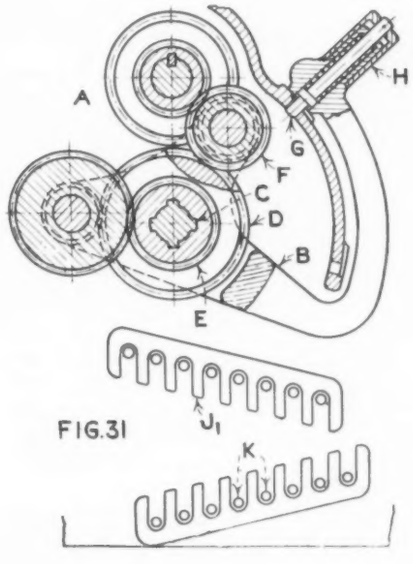


FIG. 31

How Photo-Electric Controls Are Applied—II

An analysis of the photocell equipment and control mechanisms as applied to packaging machines for positioning the wrappers

AS BROUGHT out in some typical applications of photo-electric controls described in the September number, the type of cell and relays or amplifiers selected depend on the particular requirements of the control. In applications wherein little power is needed from the control, the amplified or relayed impulse from the photocell performs the operation. But in many machines operated under photo-electric control, the power needed to perform the operation is too large to be supplied economically by the amplified output of the photocell. In each of the following typical applications the photo-electric control serves only to engage or disengage power supplied by the machine to perform the operation.

In machines designed for wrapping printed cellophane several methods are used to register automatically the printing with the cutter. In one method electrical contacts made through perforations placed in the wrapper when it is printed, close a circuit that energizes an electromagnet. The electromagnet either disengages the drive to the feed rolls or applies a brake to the cellophane strip. A cam operated by the machine opens the electrical circuit after the operation is completed and holds the circuit open until the paper starts feeding for the next cycle of operation. But electrical contacts are difficult to keep clean and are worn rapidly by the paper passing through at high speed. In addition, the hole or notch in the wrapper strip adds to the cost of the material, and may be undesirable.

If a photocell is illuminated by a light beam passing through the printed cellophane wrapper, no perforations in the wrapper and no electrical contacts are required. The International Cigar Machinery Corporation uses photo-electric control in the machine illustrated in the first article. As shown diagrammatically in Fig. 6, cigars are wrapped with cellophane on which the band has been printed. In this machine cellophane feed rolls, driven by the machine through a magnetic clutch and a set of bevel gears are designed to feed the wrapper a little faster than required. When the proper length has been supplied the photocell illumination is interrupted by the printing on the paper and the clutch is disengaged.

For the high paper speeds involved the response of the dry disk cell and its sensitive relay is too slow. Either the phototube or selenium cell is used, but the

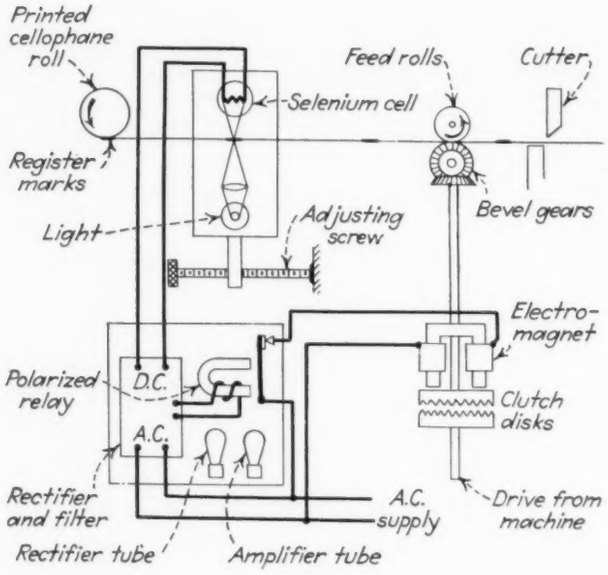


Fig. 6 — In the cigar-wrapping machine amplified power from a selenium cell operates an electromagnet disengaging the clutch that drives the feed rolls

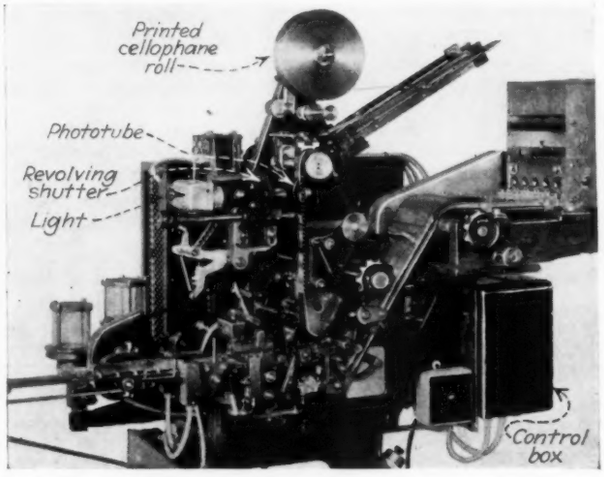


Fig. 7—Register of the printing on cellophane wrappers is under control of a photo-electric device in this machine designed by the Package Machinery Company

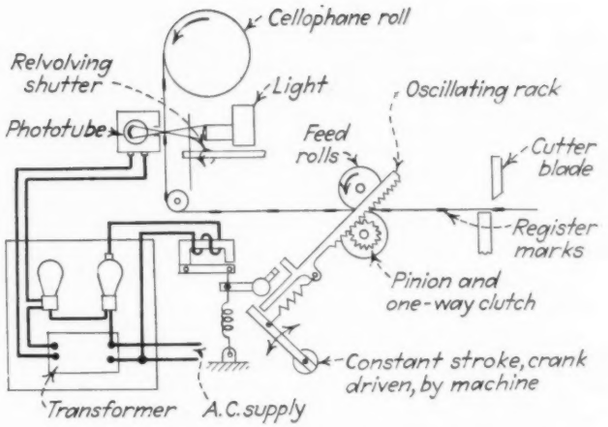


Fig. 8—When the paper becomes out of register the feed stroke is shortened by a stop moved into position by the electromagnet operated by the photocell and amplifier

selenium cell has the advantage of a greater output. Although the selenium cell does have a time lag, its response is more rapid than the action of its attendant relays so that the cell is not the controlling factor in the speed of operation. To obtain a uniform rate of operation the selenium cell is supplied with direct current by a rectifier tube and filter. An electronic amplifier, also operating on the filtered direct current, steps up the power of the cell to operate a high resistance sensitive polarized relay. Like the mercury switch previously described, this relay requires only a small direct current for its operation, and in turn controls power circuits of sufficient capacity to operate the electromagnet on the clutch. A sensitivity adjustment on the control panel varies the voltage across the photocell, and the tension of a spring on the relay is adjusted as required by the output of the amplifier.

In intermittent feed packaging machines the length of the wrapper cut off is a product of the rate of paper feed and the time of feed. As the rate of feed is increased to obtain faster machine operation, the time available for operation of the electromagnet and disengaging mechanism becomes too short to obtain reliable and accurate control of the paper length. There is available, however, a considerable time interval in each cycle of operation when the paper strip is stationary. If a fixed feed stroke is used the photocell can indicate the degree of register obtained in each wrapper after the feed stroke is completed. A small correction can then he applied to the feed stroke to bring the next wrapper into register. With this method high-speed operation of the photo-electric mechanism is not necessary and accurate registering is independent of the rate of paper feed. A control device using this principle is incorporated in a high-speed intermittent feed wrapping machine made by the Package Machinery Company.

Although the photo-electric control can be arranged to correct for register in each direction, the apparatus is necessarily complicated since two complete photocells and amplifiers will be required, or one photocell and an accurate timing device with a double set of relays and mechanical correcting devices. Practical limitations in mounting and adjusting two photocells make it difficult to provide a correction for small variations in register, and high-speed synchronized contacts are difficult to maintain. These difficulties are avoided when the normal paper feed is made a little too long, as the correction is then always in one direction, and only one cell and amplifier is needed. To further simplify the device, the photocell control is arranged to supply a fixed correction only after the paper becomes more than a predetermined allowable amount out of register. Besides involving only one photocell and one amplifier, this method requires no operation from the control device except when the paper is out of register.

This method as used by the Package Machinery Com-

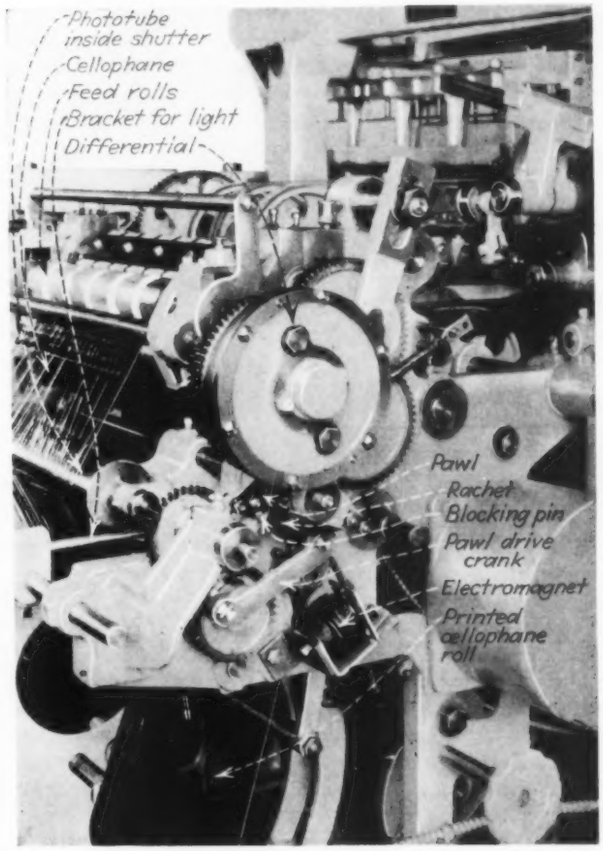


Fig. 9—For continuous paper feed a differential drive is provided to maintain register of the cellophane with the cutters

pany in the machine illustrated in Fig. 7, maintains the register correct to within $\frac{1}{32}$ in. with a wrapper length as short as 3 in. Register of the printing with respect to the end of the wrapper piece is indicated from the advancing edge of the design printed on the cellophane strip. A long slit, $\frac{1}{64}$-in. wide, in the photocell housing is accurately located at a position ahead of the cutter rolls, so that the slit is partially covered at the end of the feed stroke when the wrapper is in register.

As shown in Fig. 8, the paper feed rolls are driven through a roller ratchet by a rack and pinion. The stroke of the driving rack is adjusted so that normally a slight excess of paper is fed. The excess feed accumulates in successive cycles of operation until the paper becomes out of register and light falls on the photocell.

The electromagnet, operated by the photocell and relay tube, decreases the length of the feed stroke for the next cycle of operation by inserting a stop in the return stroke of the rack. Because of the shortened feed stroke the printing on the next wrapper piece is thus brought back into register. Since the photocell must be prevented from operating except when the paper has come to rest at the end of a feed stroke, a rotating shutter cuts off the light to the photocell during the period when the paper is advancing.

The time interval during which the paper is stationary and the photocell is illuminated is too short for reliable operation from the combination of a dry disk cell and sensitive relay. To obtain full response in the short time available either the selenium cell or the phototube with an electronic amplifier must be used. Since the high sensitivity of the ordinary selenium tube is obtained only under ample illumination, the small amount of light passing through the narrow slit in the photocell housing limits the selection to a phototube.

Because the photocell applies a fixed correction to the next cycle of operation, the exact timing of the cell impulse in relation to the machine is not important. Therefore, alternating current can be supplied to the photocell and amplifier tube from an a.c. source already available at the machine. The additional expense and maintenance involved in the use of an electronic rectifier and d.c. amplifier is unnecessary.

A single amplifier tube operating on a.c. steps up the photocell output to provide power for the operation of either grid controlled rectifier tubes or sensitive magnetic relays. With the grid controlled rectifiers or relay tube, there is no time lag in the operation and contact difficulties experienced with sensitive mechanical relays are eliminated. Being itself a rectifier, the relay tube supplies half-wave rectified direct current to the electromagnet making it necessary to select an electromagnet designed for a pulsating current supply.

In this control the relay tube has the additional advantage of operating on less power and therefore involving a lower amplification ratio. With low amplification ratios the amplifier is more stable in operation; only one adjustment, a means for varying the sensitivity of the photocell, is needed. The operation of the relay tube is unstable for a short time after current is supplied to the filament circuit. To give the relay tube time to warm up, a five minute time delay switch is automatically connected in the relay tube output circuit when the a.c. power is switched on.

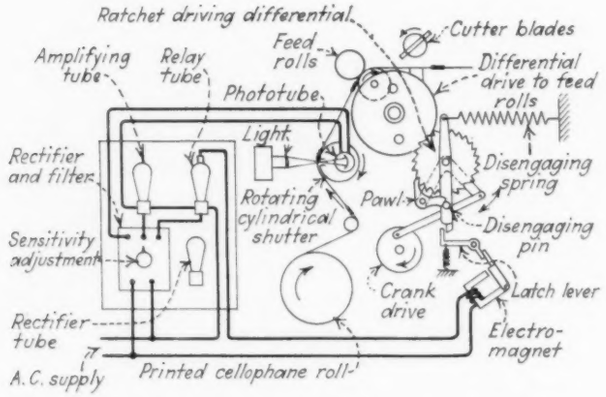


Fig. 10—No timing contacts are required when the synchronous shutter is used. Only the phototube is fast enough for the high-speed operations involved

In another packaging machine, illustrated in Fig. 9, the paper feed is continuous, presenting an entirely different problem in keeping the printing on the celloplane in register with the cutter rolls. The cutter rolls and the paper feed are driven through a differential gear arranged to shift their relative position. Manual control of the differential drive involves the same disadvantages entailed with manual control of the intermittent feed machines, with the additional factor of much higher speed of operation.

For a photo-electric control of register the power needed to shift the differential is obtained from the machine at much lower cost in equipment than from the amplified photo-electric impulse. In the design used by the Package Machinery Company, as illustrated in Fig. 10, the differential is operated by a ratchet wheel driven by a reciprocating pawl, this reciprocating motion being obtained from a crank driven by the machine. Engagement of the pawl and ratchet, controlled by the photo-electric device, requires only a small amount of power supplied to the electromagnet.

As in the machine shown in Fig. 7, the feed rolls are driven at a slightly faster rate than the cutter rolls so that the corrections applied by the photocell are only in one direction; thus, only one photocell and amplifier is needed, and only one direction of rotation is required for the ratchet drive.

With continuous paper feed, the usual method of allowing the printing to eclipse a fixed slit in the photocell housing involves a synchronous timing contact in the output of the amplifier. This timing contact is arranged so that the amplifier is disconnected from the correcting mechanism when the paper is in register, that is, when the photocell impulse comes at the proper instant the contacts are open and no correction takes place. With high paper speeds the timing device must necessarily operate with great accuracy, a condition difficult to obtain because contact points burn, wear away and accumulated dirt. But, in Fig. 10, a rotary aperture driven at the same speed as the paper strip not only does away with timer contact difficulties but also prolongs the time of exposure, allowing more light to enter and permitting the use of a narrower slit and higher paper speed.

Because of the high paper speed and the extremely short period of illumination on the photocell, the high-speed combination of a photo-emissive tube and the relay tube is necessary. The duration of the light flash is so short that with a.c. operation as used for other devices, the light flash may occur during an inoperative period of the cell. An electronic rectifier and filter is therefore required to supply d.c. to the phototube and amplifying circuit.

In the use of the relay tube in this control there is an advantage in using d.c. supply for the electro-magnet. When controlling a d.c. circuit, the operation of the relay tube is continuous once the discharge is started. An impulse of extremely short duration, even less than 0.001 sec., starts the operation of the tube, the operation continuing until the discharge circuit is interrupted at some point outside of the tube. Thus, the operation of the electromagnet, started by an extremely short light flash, continues until a cam operated by the machine opens the circuit after the correction is completed.

Ball Bearing Mountings for High-Speed Spindles—II

Conclusion of the article begun in September discusses straight roller bearings, lubrication and locking devices

F. A. FIRNHABER, *Designing Engineer, Landis Machine Company*

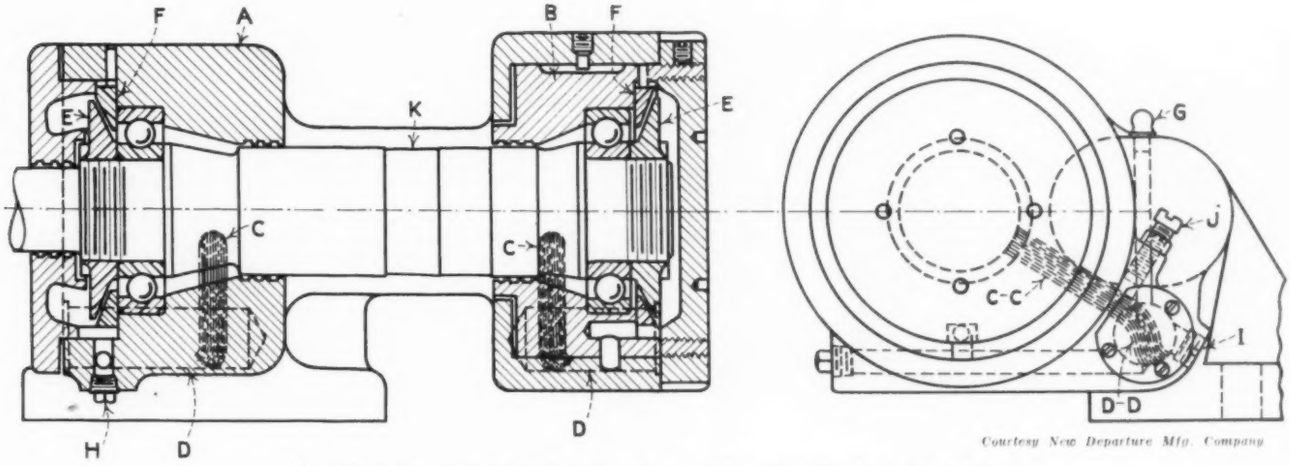
MANUAL adjustment of the end play in high-speed spindle bearings is sometimes desirable in order to preload the ball bearings. In a design shown in Fig. 9 in the preceding article, the axial load was adjusted by turning a nut bearing against the outer race of one of the ball bearings.

Another design wherein the thrust load on the ball bearings is adjusted manually is shown in Fig. 10. In this construction both the outer and inner race of each of the bearings is assembled with a press fit in the housing and on the shaft respectively. The outer race of the front bearing is a press fit in the heavy housing frame *A*, while the outer race of the rear bearing is a press fit in the heavy adjustable sleeve *B*. This sleeve is threaded into the nut shown at the extreme right end of the longitudinal section. The position of the nut on the collar *B* is fixed by a setscrew as shown. Loosening the setscrew and turning the nut pulls the adjustable sleeve *B* to the right. In this manner a sufficient thrust load is applied manually to bring the balls of both bearings into positive contact with the races.

It will be noticed that it is only the fixed position of the left-hand bearing that prevents the sleeve *B* from moving to the right. The reason for this construction is to permit expansion of the spindle longitudinally. As the spindle elongates, on account of being heated, the thrust load on the ball bearings is relieved.

Lubrication of the bearings in this design is quite similar to that used in the design of the mountings shown in Fig. 9. Lubricant passes through the wicks *C* and *C* to each of the bearings, being drawn up from the reservoirs *D* and *D* which are drilled and capped holes. The wicks contact the tapered surfaces of the spindle, the oil creeps to the highest point of the taper, and is thrown in a fog or mist into the ball cage and raceways. To accelerate this action, the rotating nuts *E* and *E* and the stationary flanges *F* and *F* are designed so as to cause a pumping or suction action, thereby drawing the mist of oil into the ball cages, through the bearings and into the return hole leading back to the reservoirs. The reservoirs are filled at *G* and drained at *H* and *I*. By means of the dog-pointed setscrew *J*, which is provided with a locknut, the wicks can be compressed to retard the flow of oil.

Both housings have tapered bores at the rear of each bearing, thus draining oil into the bearings when the spindle is brought to a stop. This assures immediate lubrication when the spindle is started again. Other-



Courtesy New Departure Mfg. Company

Fig. 10—Preloading in this design is partially relieved when the spindle warms up and expands. A small amount of oil is retained at the bearings to provide lubrication when the spindle is started

wise there might be a deficiency of lubrication until the oil has been warmed sufficiently for the capillary attraction to feed the lubricant effectively.

Three unusual designs of spindle mountings are shown in Figs. 11, 12 and 13. These spindles were designed and built by Ex-Cell-O Aircraft & Tool Corporation and are mounted on bearings of their own manufacture. These double-row-type bearings have ball retainers of a fabric-base bakelite material. It is claimed that the retainers take on a high polish when lubricated properly, thereby reducing wear. Also, if particles are dislodged from the retainers they will not injure the bearings as the material is of a non-abrasive character.

The spindle shown in Fig. 11 runs at 10,000 r.p.m. The outer rings of the bearings are clamped against the housing shoulders *A* and *A* by the inner one of the locking nuts *B*. The outer locking nut *B* serves as a locknut. Between the two nuts is placed a felt ring which seals the bearings. The bearings are preloaded, the abutting faces of the outer races of the bearings having been ground so that when the locknuts *B* and *B* are tightened the bearings will be loaded axially to the required amount.

Both the slinger and the left-hand end bearing are clamped to the shaft with double locknuts *C* and *C*. The bearings on each side of the pulley are keyed to the shaft at *D* and *D*, the two keys being placed diametrically opposite to each other to retain balance. A taper pin holds the pulley in place. Lubrication is furnished through sight feed oilers.

The same general type of bearing mounting is used in the two-unit spindle shown in Fig. 12. The two units are connected by a flexible coupling *A*, which not only drives unit *B* from unit *C*, but also dissipates any vibration that may be introduced to the pulley driving unit because of unevenness of the driving belt. Primarily, the reason for using a two-unit spindle is to make it possible to have the bearings close together, as a long span between bearings might cause whipping of the shaft.

In Fig. 13 is shown the design of a spindle for operating at a speed of 65,000 r.p.m. This is used for solid-stick or pencil grinding wheels whose hole-grinding capacity is from $\frac{3}{32}$ to $\frac{3}{16}$ in. diameters. The grinding wheels are held in a chuck whose shank is screwed into the hole A_1 of the spindle *A*. At such high speeds it is not permissible to run the risk of vibrations from uneven belts or improperly balanced motors, consequently this spindle is driven by compressed air.

The bearings for this spindle are of the same type and are mounted in the same manner as in Figs. 11 and 12. At the rear end of the spindle is the turbine wheel *B*, which is driven through the air inlets *C* and C_1. Four equally spaced channels, *D* and D_1, are located in the double steel bearing shell which is $1\frac{5}{16}$-in. outside diameter. The exhaust air from the turbine rushes through these channels and carries away the heat generated at the bearings because of the extremely high speed. As the ribs separating the channels are narrow, a large percentage of the surface of the bearing housing proper is exposed to the blast of the exhaust. Incidentally, this exhaust blast also serves the added purpose of preventing the entrance of grit into the bearings.

As the air furnished to the turbine must be clean and dry, the airline is provided with an air strainer and moisture trap, not shown in the drawing. An-

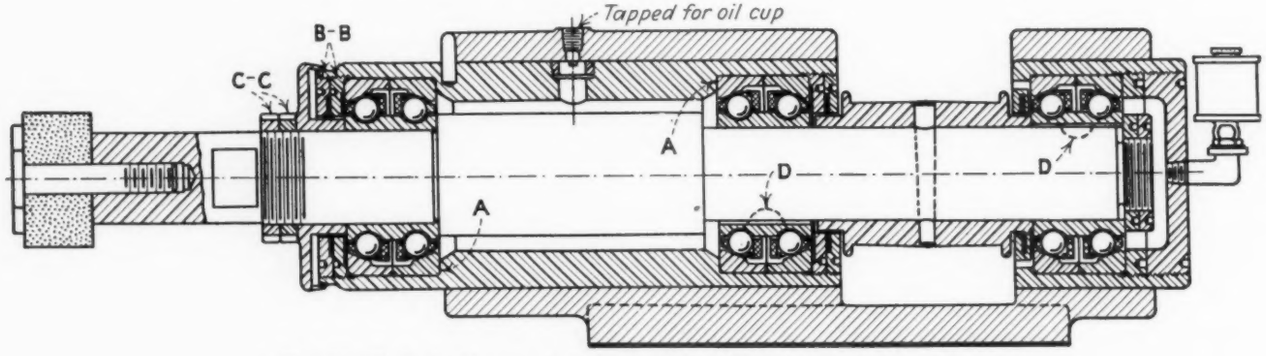


Fig. 11—Each double row ball bearing has fabric base bakelite retaining rings. Bearings are ground so that the locknuts preload each bearing

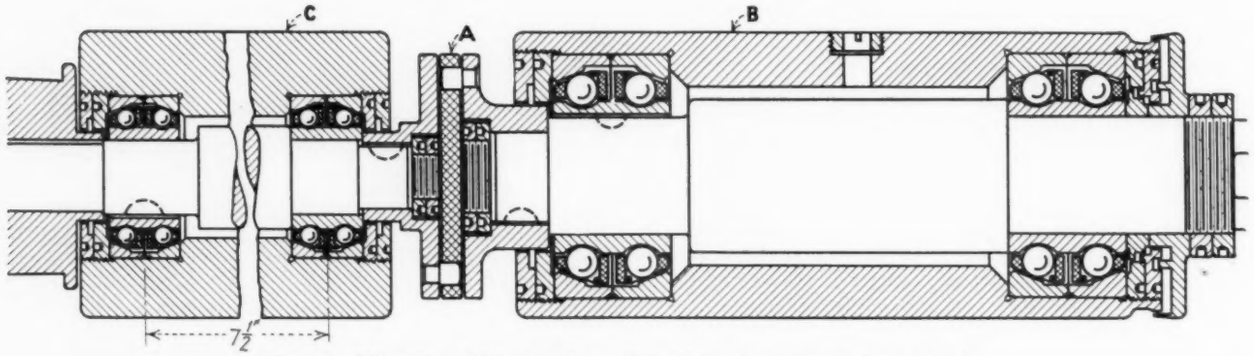


Fig. 12—Vibrations arising in the pulley drive are prevented from entering the right-hand shaft by a flexible coupling

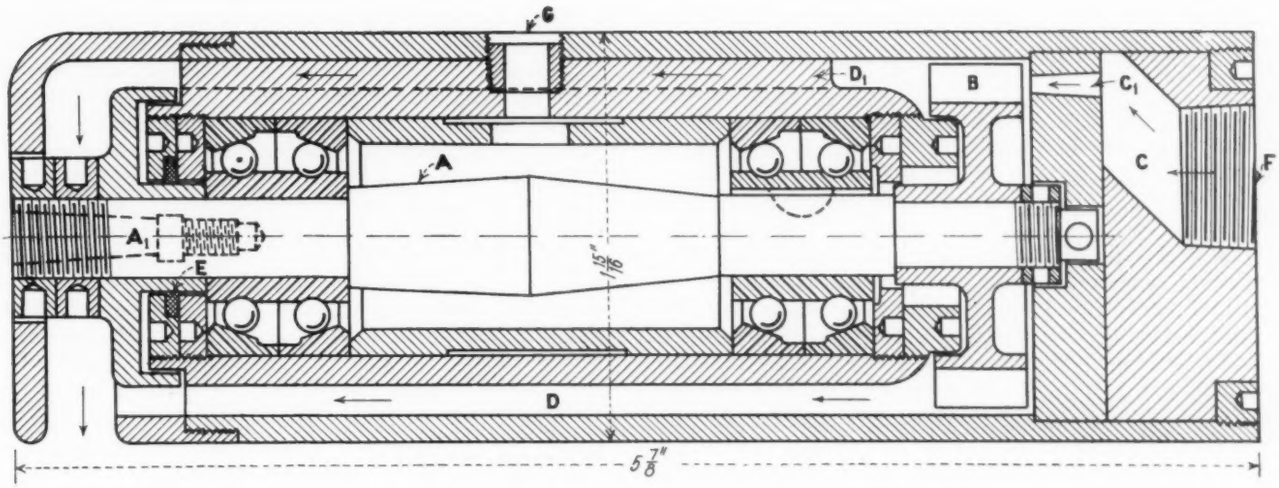


Fig. 13—In addition to supplying vibrationless power for high-spindle speeds, a compressed air turbine drive supplies a cooling air blast and keeps dirt out of the bearings

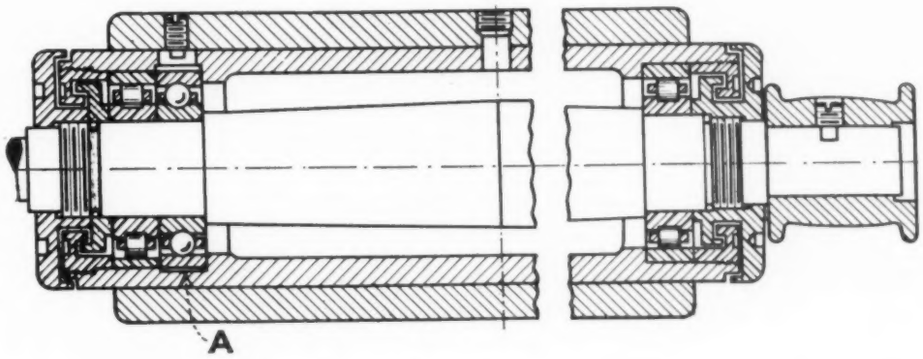


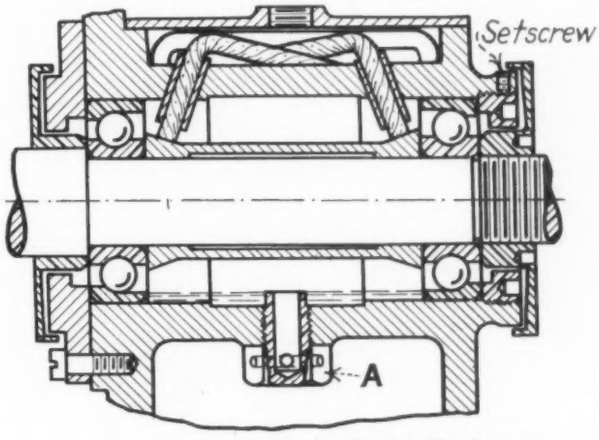
Fig. 14—Roller bearings can carry greater radial loads than ball bearings, but one ball bearing is required to carry the axial load

other interesting feature is a speed gage, calibrated in thousands of revolutions per minute, which is mounted on the airline at *F*. This gage indicates the speed at all times. Speed is regulated by the adjustment of an air throttle. With 90 lb. per sq.in. air pressure, the air consumption is slightly more than 15 cu.ft. per min.

Until recently, straight roller bearings were considered unsuitable for extremely high speeds. However, in the spindle mounting shown in Fig. 14, Norma-Hoffmann ball and roller bearings are used and spindles of this design have served successfully at speeds as high as 35,000 r.p.m. Compared with ball bearings, size for size, the straight roller bearing has a greater radial load-carrying capacity but can take no end thrust whatever. Therefore, at the front end of the spindle shown in Fig. 14, a ball bearing is introduced to take care of the axial thrust. The sole function of the ball bearing is to carry the thrust loads. Any attempt to have this bearing carry radial loads would require not only a high degree of accuracy in the machining but would also possibly result in the ball bearing carrying all the radial load instead of the roller bearing. Therefore, a definite clearance is given between the outer race of the ball bearing and the housing, as shown at *A*. Thus, the roller bearings carry all the radial load and none of the thrust, while the ball bearing carries all the thrust load and none of the radial load.

In applications where the speeds are extremely high, the oil reservoir may be located above the bearings as in Fig. 15 in order to utilize gravity flow in the wicks. Care must be taken in all such mountings to adjust the wicks so that only an extremely light contact is made with the slinger. Otherwise, the wicks might become charred. The charred end will gradually disintegrate and form an abrasive which will attack the finely polished surfaces of the bearing parts. Furthermore, the charring will retard the capillary action of the oil, ultimately causing failure of the bearing.

After the oil mist has passed through the bearing it drains into a shallow reservoir. The oil level in the reservoir is kept below the bottom of the balls in the bearing, the excess oil flowing out through the combination breather and drain plug. If such an overflow



Courtesy New Departure Mfg. Company

Fig. 15—Oil fed by the wicks from the reservoir above the bearing drains into the sump and out through the drain plug

is undesirable, the excess oil can be drained into a common sump on well, located below the bearing and drained at will.

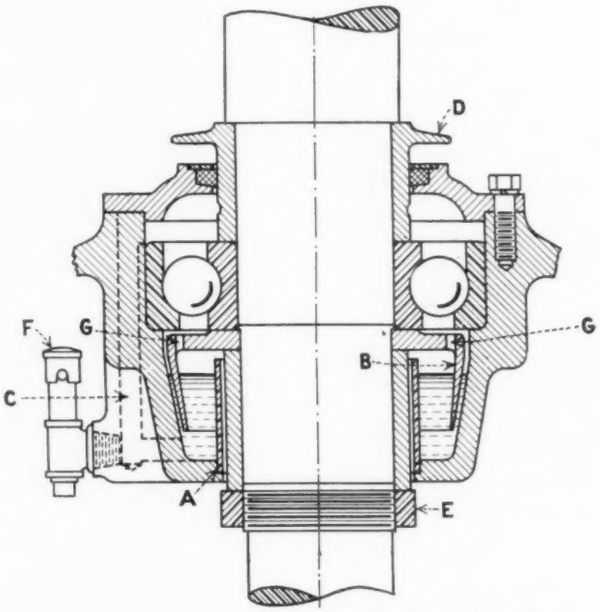
Since lubrication is such an important factor in the design of high-speed spindle mountings, several additional typical methods of providing lubrication will be described. In Fig. 16 is shown a construction in which the oil pocket is directly below the bearing. This is not only an economical construction but also permits outside adjustment of the wick. In this design the wick contacts a short slinger *A*, which is pressed on the shaft between the bearing and the shoulder of the shaft. The slinger action is as described previously. After the mist of oil has passed through the bearing on the left side, it is collected and led back to the reservoir through groove *B* and hole *C*. Any oil that might tend to leak out at the right end of the bearing housing is collected in groove *D*. The oil creeping along the shaft reaches groove *E*, the sharp edges of which throw the oil off by centrifugal force into groove *D* from which the oil is drained through hole *F* and thus back into the reservoir.

In vertical spindle mountings, the lubricating problem is intensified, since the question of leakage is added to that of proper feed. Two successful applications are shown in Figs. 17 and 18. In Fig. 17 the quill *A* has oil-tight joints in the bearing housing. This prevents leakage and a suitable oil level can be maintained below the bearing. The tapered slinger *B* is clamped to the shaft and sprays or atomizes the oil into the bearing. Oil that passes through the bearing is drained back into the lower chamber through the hole *C*. This method not only prevents accumulation of oil in the bearing but also provides a constant finely broken up spray which is thrown into the bearing in such a manner as to furnish positive lubrication of the ball separator pockets.

It will be noted that the bulk of the lubricant carried upward and into the bearing by the slinger *B* must pass through a series of apertures *G* and *G* before striking the bearing. It is the feature of this design that the size and number of apertures can be changed to suit the requirements of various installations.

Slinger *D* is provided with a felt seal to prevent dirt entering the bearing from above. One nut *E* clamps all the assembled parts. The oiler *F* maintains the proper oil level by providing an overflow.

In Fig. 18 a wick is used which extends across and lightly contacts with the upper slinger *A*, which is clamped above the bearing. It is typical of all designs of this character, wherein the oil is drawn from a source below the bearing, that it is extremely essential that the wick shall actually make contact with the slinger. If actual contact is not made a sufficient flow of oil will not be induced. In the design shown here the slinger *B* below the bearing prevents the oil from running down the shaft. The spiral groove or thread *C*, which is turned in the wall of the bearing housing *D*, is similar to the construction explained under Fig. 3, but there is another feature to be considered. In this design the groove is cut into the non-rotating member. The groove being stationary, there will be



Courtesy New Departure Mfg. Company

Fig. 17—In this vertical shaft a slinger running in the oil reservoir throws a spray of oil into the bearing

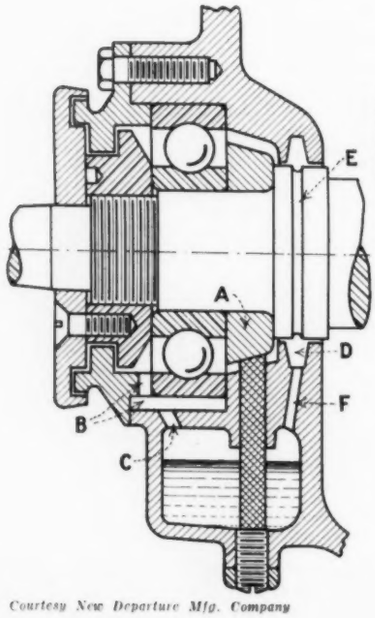


Fig. 16—Outside adjustment of the wick is provided in this housing having an oil reservoir below the bearing

Courtesy New Departure Mfg. Company

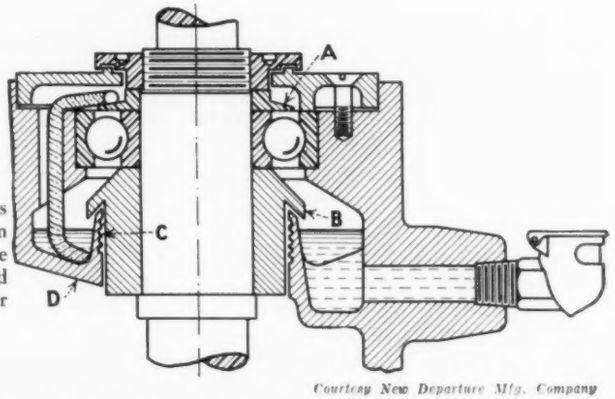


Fig. 18—When oil is drawn by a wick from below the bearing the wick must make good contact with the slinger

Courtesy New Departure Mfg. Company

no tendency to throw off the dirt and hence the groove is likely to become clogged with dirt if the unit is placed in an unprotected position.

In Fig. 19 is shown another type of wick oiler for a horizontal shaft. The spiral spring *C* is wound small at one end to grip the wick and is inserted in a cartridge or capsule as shown. The capsule is screwed in to press the wick lightly against the slinger. Constant contact is assured. Here also, the slinger is provided with a shoulder to facilitate pressing the bearing off the shaft if necessary.

Particularly in high-speed work, the design of even the minor details such as locknuts and washers is of

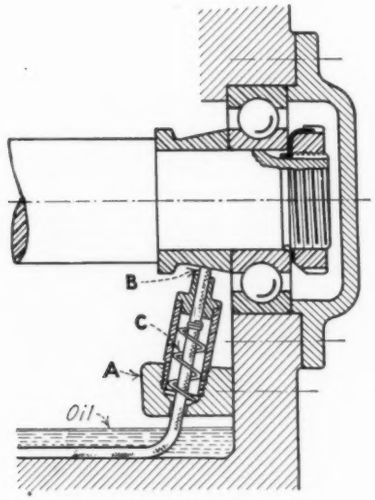
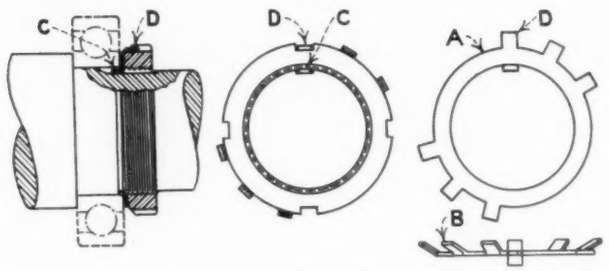


Fig. 19—Light contact is maintained between the wick and slinger by the spring

Fig. 20—Consisting of a slotted locknut and lock washer, this standard product gives 24 locking positions in each revolution



extreme importance. In Fig. 20 is shown the design of a locknut and washer which is particularly adaptable to high-speed work. Balance is maintained and there are 24 locking positions for each thread pitch. This type of locknut and washer eliminates the necessity of an additional locking nut. Also, it is never necessary to use a left-hand thread as is required when using a plain nut and locknut for certain directions of rotation wherein the tendency is for the nut to be rotated counter-clockwise.

At *A* and *B* in Fig. 20 are shown the details of the washer. The shaft is slotted thread deep at *C*, as indicated in the assembly view, to accommodate the projecting key *C*. After the nut has been screwed in place, a coinciding prong *D* of the washer is bent into engagement with a slot in the nut. The remaining prongs are left as they are to rest against the angular surface of the nut. Sometimes these prongs act as a slinger. As this type of locknut and washer is a standard product and is carried in stock by all bearing manufacturers, they are not only economical but are also readily available.

After Designing—What?

C. F. Cowdry, Jr.
Cowdry Machine Works

"After the laboratory or the engineering department has completed its job, there is nothing to it; all you have to do is to go ahead and make it according to the drawings." Such is the opinion held by many people, some engineers included, of the job of the manufacturer of machinery who undertakes to build a machine or product according to the plans and specifications furnished by the customer. It is assumed that the drawings are perfect and nothing has been omitted.

But it is simply humanly impossible, regardless of the individual's experience and training, so to perfect a set of drawings that in the first article made from them no changes will be necessary. If all designers would only be willing to admit that the machinery builder and his engineers are able to make worth while suggestions for changes, and even improvements in his drawings, much less experimentation and expense in the machine itself would be necessary.

To have a correct set of drawings is, then, the machine builders' first job. The next step of building a model is found necessary in some work but not in others. Sometimes this is a simple wooden affair made in the pattern shop; and again it may be a more elaborate model made of material similar to that going into the ultimate machine. In some instances we need only to prove a certain part of the mechanism, at other times a real working model of the machine to be built is required.

The customer's next logical question is, "How much will it cost?" Usually this is followed by one not so logical, namely, "and will you guarantee that it will work to my satisfaction?" With a complete and detailed set of drawings and with the knowledge that the principle of the machine is correct, the manufacturer can estimate the cost. But he can not, and should not, guarantee that it will work to the customer's satisfaction. He can state that the machine will work, that it will produce the article desired or perform the service required, and do this at a certain rate, but he can not guarantee that when the machine is completed the customer will be satisfied.

It would be quite natural to assume that, provided the first machine did its duty, the manufacturer of it would be the one, or at least among those to be asked to build additional machines. Unfortunately, however, in some instances this is not the case, simply because there are people who fail to realize that the cost of the first machine is always high, and of course very much higher than it will be for subsequent ones. When the customer does not realize this and he sees this difference in cost from, say some outside estimate, he may turn directly away from the builder of the first machine and not even give him the opportunity to show that he, too, can build the subsequent machines at an appreciably lower figure.

Happily, though, the customer usually stays with the concern that built his first machine, and thereby permits it to carry through to a conclusion the functions that it is capable of performing.

Selecting the Motor To Fit the Requirements—II

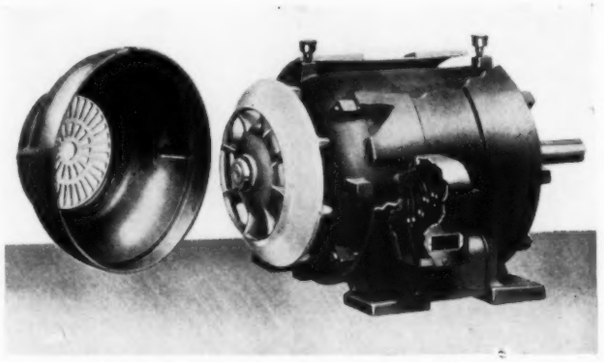
A discussion of motor ventilation and the effect of non-uniform air gaps on noise and vibration

THE maximum temperature at which a motor can operate is quite definitely limited by the temperature that the insulation can withstand without deterioration. Most of the heat generated in a motor is the result of the internal electrical resistance of the windings and the core losses caused by eddy currents and hysteresis. The internal electrical losses will increase with the horsepower output of the motor. But for a given rate of heat generation, the maximum temperature rise in the motor will be limited by the heat carried away by ventilation. This last will be greatly affected not only by the volume of cooling air, but also by the mechanical design of the motor. Frequently, particularly in the larger sizes of motors, ducts are provided in the stator or rotor or both. Such ducts not only increase the cooling surface of the motor, but they are designed so that the path of heat conduction from the hottest part of the motor to the cooled surface will be of minimum length.

Usually the allowable temperature rise for motors is specified as 40 deg. C. above the surrounding temperature. It is possible to obtain motors having an allowable temperature rise of 50 deg. C., the windings of which have special insulation. An even higher temperature rise is permissible with certain motors of highly specialized design. Such motors, however, are relatively expensive.

The ordinary open-frame type motor depends solely upon a natural circulation of air to carry off the heat. By placing a fan on the motor shaft the heat carried away can be greatly increased, thereby making it possible to obtain a higher horsepower rating from the same motor. Forced ventilation is also used for all inclosed motors and most built-in motors and for all conditions where natural circulation is restricted. This phase of the ventilation problem is always handled by the designers of the motors, the motor manufacturer specifying the cubic feet of air per minute that must be available to the motor to secure the required ventilation. This information is often necessary in order that the designer can calculate the size of the openings required in the housing in which the motor might be placed.

Perhaps one of the most common mistakes made by designers when installing motors in the base of the driven machine is that of failing to provide proper openings for ventilation of the housing. Regardless of the type of motor and the size of the space in the base of the machine, louvers or openings must usually be provided. Only when the motor operation is extremely intermittent may the louvers be omitted. If the housing is closed entirely, the air merely recirculates through the motor. The heat dissipated is then limited to the conduction and radiation losses from the housing. The temperature of the air within the housing will increase until an equilibrium has been established between the motor heat losses and the heat losses from the surface of the housing.



Courtesy Allis-Chalmers Mfg. Company

A typical construction of a totally-inclosed fan-cooled motor, showing how air is blown through the annual space between the outer frame and the inclosure

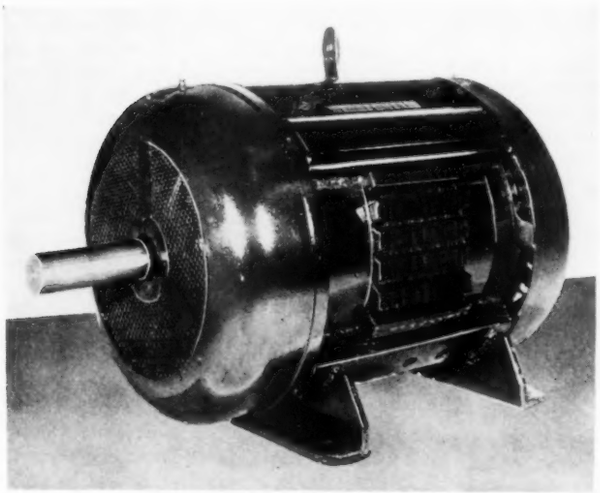
Openings in the housing to admit cool air and permit the heated air to go out should be placed so that there will be a natural circulation. With one set of louvers placed near the bottom of the housing and another set near the top, there will be a chimney effect. The position of the motor inside the housing should be such that the motor will be in the path of the natural circulation of air.

When the motor is provided with a fan mounted on the armature shaft, the fan will create a blast of air in a definite direction. The best arrangement is then to have a passage through which the air will be forced to travel, thereby preventing the recirculation of heated air. This can be accomplished by designing the housing so that there will be a duct or passage leading from holes in the housing to the motor. This passage may be either for the intake air or the heated exhaust air, preferably the latter.

Although the permissible temperature rise in motors is specified as a definite number of degrees above the surrounding temperature, this only obtains when normal ambient temperatures exist. It must be remem-

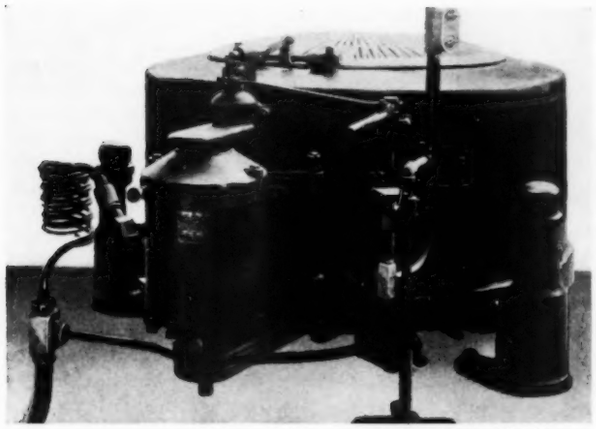
bered that the temperature to which the insulation is subjected is the primary factor limiting the rating of the motor. Therefore, if the motor is required to operate in an atmosphere of high temperature, its horsepower rating will necessarily be lower.

Consideration must also be given to the possibility of contamination in the atmosphere surrounding the motor. Sawdust in woodworking factories, lint in textile plants, corrosive fumes in chemical plants and air heavily laden with dust, as in foundries, are examples of the type of air contamination frequently met. Special provisions must be made for motors operating under such conditions. For severe conditions, totally inclosed motors are used, and cooling air is furnished to the motor through a duct leading from the motor to some point where uncontaminated air is available. The fan on the motor shaft provides the necessary circulation. In some instances, a second duct leading from the motor to a distant point is also provided for the heated air expelled from the motor. Another method is to use a separate blower to ventilate a number of motors, ducts leading from the blower to the individual separate motors.



Courtesy Wagner Electric Corporation

Motor provided with slots in the stator laminations in order to increase the cooling area. This motor is supplied with a fan at each end of the armature



Courtesy Louis Allis Company

Totally-inclosed motor for driving a centrifugal dryer. To permit adjustment of the V-belt tension, the motor is mounted on a hinge

It is not always necessary to provide uncontaminated air for ventilation. Many designs of inclosed motors, such as the one illustrated here, are available. In such motors the ventilating air does not come in direct contact with any of the motor parts other than the frame. The strength of the air blast created by the fan is such that there is little or no opportunity for dust or lint to clog the ventilating passages.

With the increasing importance being attached to the absence of noise and vibration, motor manufacturers have been putting much effort on the design of machines that will be free from noise and vibration. This pertains not only to motors for household equipment, office appliances and similar applications, but also to larger motors. Most of the noise developed in an electric motor arises from electrical hum, unbalanced magnetic fields and bearing noises. In addition to reducing these noises to a minimum, resilient motor mountings are sometimes applied to confine the vibration to the motor itself.

When the machine manufacturer purchases the motor parts and assembles them into the housing of his machine, care must be taken to avoid a non-uniform air gap between the rotor or stator. The bore of the hole for the stator housing must be concentric with the rotor shaft bearing. In addition, also, the housing bore for the stator must not be out of round. Finally, the diameter of the bore must be correct, so that there will be neither play between the stator and its housing, nor too great a pressure required for assembling the stator into its housing. This might cause the stator to become distorted, thereby resulting in a non-uniform air gap. This also results when the bore of the stator housing is out of round or is too large.

A non-uniform air gap will unbalance the magnetic field. The starting torque will be uneven and the unbalanced magnetic field will set up noises. The unbalanced pull on the armature may cause bending of the shaft, increased bearing pressure, or may alter the location of maximum bearing pressure. This last might cause trouble in lubrication, as it might throw the point of maximum oil film pressure close to the oil hole. Under such conditions, the lubricant would be pumped out of the bearing.

The air gap between the rotor and the stator is usually between 0.01 in. and 0.02 in. The maximum allowable variation in the air gap should be limited to not more than 0.002 in. That is, the maximum eccentricity between the bore of the stator and the rotor shaft should not exceed 0.001 in. If the rotor is sufficiently off center to cause mechanical rubbing between the rotor and the stator, the motor will heat up rapidly and a great amount of power will be wasted as heat. This power loss is not only the result of mechanical rubbing, but is accentuated by the magnetic clinging between the surfaces of the rotor and stator.

A light press fit is considered the best way of mounting the stator in the bore. Some motor manufacturers put a rolled steel frame around the stator, this rolled steel frame being machined for the end bell and machined on the outer surface in order to assure the correct diameter for the stator assembly. Usually, however, the frame is omitted.

COMMUNICATION and COMMENT

Shear Stress in Machine Elements

To the Editor:

In the September *Product Engineering*, P. N. Bland questions the statement that the principal shear stress is the criterion of failure in structural materials. The facts as we know them are as follows. Failure from stresses in a structural part always starts at the most highly stressed section when the material has reached its yield point in shear. This fact has been verified time and again by microscopic metallurgical analysis of the overstressed grain structure. This fact is as true for simple tension and simple compression as for the most complicated stress distributions. In tensile deformation, the simplest case of stress distribution, both axial stress and lateral contraction occur. The tensile deformation is entirely due to the shear stresses on planes inclined to the tensile stress, and the shear strength of the material fixes the elastic limit and to some extent the ultimate strength of the material in tension.

The confusion that exists is largely brought about by the fact that when we talk about failures, many engineers have in mind the ultimate failure of the material. In all structural parts elastic failure begins when the material has reached its yield point at the most highly stressed section. When the materials are subjected to shock or fatigue stresses, initial failure may rapidly develop into ultimate or complete failure. In the article on photoelasticity the term failure refers to elastic or yield point failure.

For loads at or near the ultimate failure the maximum shear stress cannot be determined on account of the local contraction which takes place at and near the zone of failure. Handbook formulas are based on values obtained under stress conditions which exist at or near ultimate failure, and are at best only roughly representative of the actual stress distributions.

For engineering purposes the laws governing the yield point or elastic failure are of the most importance and are now fairly well understood. More detailed information on the subject is given in the following list of references:

"Code and Design of Transmission Shafting," Engineering and Industrial Standard of the American Standards Association, pp. 16-19.

"The Theory of Strength," by A. Nadai. *Applied Mechanics*, American Society of Mechanical Engineers. July-September, 1933, p. 111.

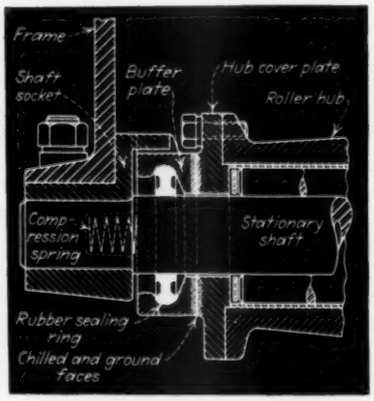
"Working Stresses," by C. Richard Soderberg. *Applied Mechanics*, American Society of Mechanical Engineers, July-September, 1933, p. 131.

—Albin H. Beyer
Department of Civil Engineering
Columbia University

Bearing Seals in Track Rollers

A. E. Palmer
Morristown, N. J.

Because of practical limitations in the design of track rollers for crawler-type tractors the roller diameter is necessarily small and the center is therefore close to the ground. Because of the conditions under which crawler tractors operate, the track roller bearings are subjected to mud and water, and if this abrasive matter enters the bearing rapid wear naturally follows. A satisfactory seal for track roller bearings must exclude water and grit, retain the bearing lubricant, compensate for wear of bearing and seal parts and, in addition, the seal must be sturdy in design, of simple construction and low cost, using a material unaffected by water.



Ground joint between buffer plate and roller hub provides a seal against mud and water unaffected by radial wear

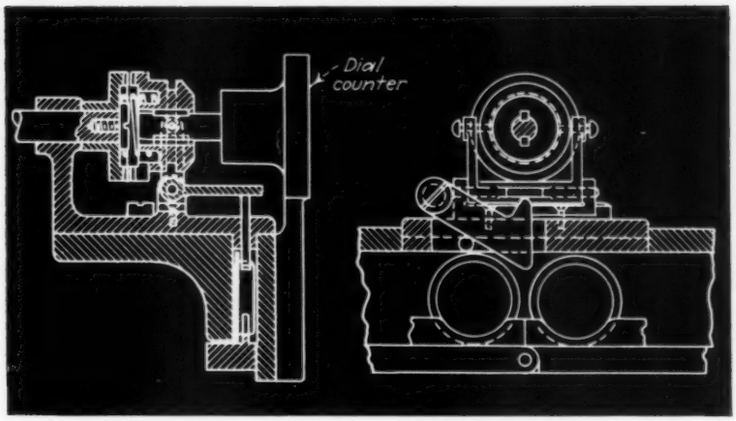
Many types of seals have been developed, involving the use of materials such as leather, cork, felt and various combinations. Some of these materials are satisfactory until wear occurs between the parts, and then the radial wear and eccentricity between the rotating parts destroys the effective seal. In the bearing seal shown in the illustration foreign matter is kept out of the bearing inclosure by a flat ground joint unaffected by radial wear of the bearing or shaft.

As shown in the illustration a buffer plate, supported on the stationary shaft by a sealing ring of grease-resisting soft rubber, bears against a plate attached to the end of the roller hub. These rubbing surfaces are chilled and ground, and are held in contact by two compression springs. Rotation of the buffer plate is prevented by engagement of a lug in a slot in the shaft socket, allowing the buffer plate considerable radial freedom. The section of the sealing ring is reduced at the center to allow displacement of the buffer plate along the shaft without the sealing ring sliding on the shaft.

Counting Device

Vincent Waitkus
Baltimore, Md.

Mechanical counting devices provide a means of determining total production, but when the product is of frail construction the pressure required by usual counting devices may damage the product. An example is found in the device shown in the accompanying illustration for counting small bottle caps. Since the count had to be made exact, a positive drive was necessary yet an exact count was obtained without damage to the product. Power to operate the counter is not supplied by the cap itself but by the shaft extending to the left, driven at constant speed. The cap feeler shown in the right-hand sketch rides over the caps as they pass on a constant-speed conveyor. When



In this counting device the power to operate the counter is not supplied by the article being counted

passing over a cap, the feeler is lifted and engages a toothed clutch, connecting a dial counter to the constant-speed drive shaft. Since the caps move past at uniform speed, the time of clutch engagement is constant for each cap, and the dial registers an accurate count.

Solutions to the Gear Problem

Here are three more analytical solutions to the gear problem submitted by C. A. Wiken in the August number. As in the solutions published in the September number, the use of these methods involves long drawn calculations in obtaining the radius and location of the common drive gear. Comparison of the various sets of dimensions submitted shows no general agreement for either the third or fourth decimal place, indicating the difficulty involved in carrying through the calculations required for a general solution.

To the Editor:

The following analytical solution involves a large amount of work as the requirement of accuracy to the fourth decimal place necessitates working to the eighth decimal place when squaring and obtaining square roots. The first step in the solution is to find by ordinary trigonometry the coordinates of the points A and C. Then by the formula for the distance between two points the distances from D to A, B and C are found as follows:

$$BD = R + 1.4 = \sqrt{x^2 + y^2} \quad \ldots\ldots (1)$$

$$AD = R + 0.8 = \sqrt{(x - x_1)^2 + (y - y_1)^2} \quad \ldots\ldots (2)$$

$$DC = R + 1.1 = \sqrt{(x - x_2)^2 + (y - y_2)^2} \quad \ldots\ldots (3)$$

Squaring Equation (1) and subtracting from Equations (2) and (3) squared, gives two new equations containing only first powers of x and y. The value of x in terms of y is:

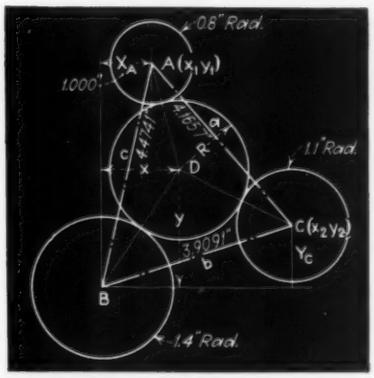
$$x = 0.3031y + 0.8329 \quad \ldots\ldots (4)$$

Comparing Equations (1) and (2) when arranged in terms of R, gives another relation involving only x and y. Rationalizing this relation by squaring both sides twice and substituting the value of x from Equation (4) results in the following second power equation containing only y:

$$21.3598y^2 - 84.0951y + 80.6756 = 0$$

Solving for x, y and R from these relations gives $x = 1.5245$, $y = 2.2818$ and $R = 1.3442$ for the dimensions required in the problem.

—Dean E. Hobart,
Ann Arbor, Mich.

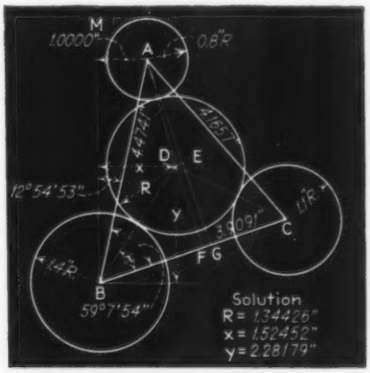


In Mr. Hobart's analytical solution, to obtain the required accuracy involves working to eight decimal places

To the Editor:

Although it is possible to use a direct method of solution to obtain the desired values, the self-checking method given below is not as laborious and lengthy, and in addition, reduces the opportunity for making numerical errors.

First, graphically determine the approximate radius of the mongrel gear, correct to within 2 or 3 per cent, either by the construction on page 104 of Marks' handbook, second edition, or by merely trying various settings of a compass until a suitable radius is found. For this problem the approximation is 1.35 in. Then draw the line AG perpendicular to BC. By the sine of half the angle formula the angle ABC is 50 deg. 7 min. 54 sec. In the triangle ABG, AG is then 3.8404 in. and BG is 2.2955. Using 1.35 in., the approximate value of the radius, the distances BF and DF are found and from these the distance AD. By averaging the value of the radius obtained from AD with the approximate radius, a new and more correct value for the radius is secured. The angle DBC by the sine of half the angle formula is found to be 38 deg. 28 min., giving DF equal to 1.710 in. and BF equal to 2.153 in. Now DE equals FG which equals BG minus BF, and AE equals AG minus EG, EG being equal to DF which is known. Thus AD in triangle ADE is found by applying the formula for the length of the hypothenuse. The distance AD is found to be 2.135 in. and the radius R' is 1.335 in.



An approximate solution submitted by Mr. Wickstrand can be carried out to the required accuracy without laborious computations

The original value for R was 1.35 in., which now appears to be too big. By averaging the radii to each of the three mating gears a new and better trial radius is obtained, namely, 1.345 in.

Repeating the process using the value of 1.345 in. instead of 1.35 in. for the radius a still more accurate value is obtained. This process is repeated until a value of R' is obtained that does not differ from the trial value of R by more than 0.0001 in. This method is not as long as it appears since the calculations can be arranged in compact form, omitting the algebraic equations, performing only the numerical work. Some of the

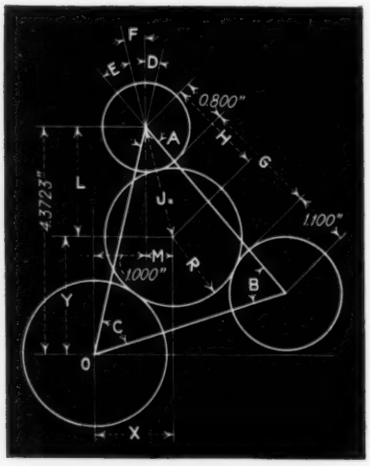
logarithms have to be looked up only once and the value of DE is so small that it only has to be found for about every second or third trial. The method has the advantage of being self checking since each new trial forms a check on the previous one.

Using this method the radius is 1.34426 in. Angle ABM is 12 deg. 54 min. 53 sec., and the angle DBC is found to be 38 deg. 17 min. 58 sec. in the final trial. Thus the angle DBM is found to be 33 deg. 44 min. 49 sec., and the coordinates of the gear center are $x = 1.52452$ in. and $y = 2.28179$ in.

—NORMAN M. WICKSTRAND,
Meriden, Conn.

To the Editor:

First the angles, A, B and C, are found by the sine relation as follows: $A = 53$ deg. 27 min., $B = 66$ deg. 51 min., $C = 57$ deg. 42 min. Angle D from the data given in the problem is 12 deg. 55 min. Angle F is found to be 13 deg. 48 in. 30 sec. by subtracting from A the angle D and the



Although not a general solution, Mr. Strom claims his method obtains the dimensions required

angle E, which is one-half of A. Dimension G is 2.2657 in., and the dimension H is therefore one-half of G plus 0.800 or 1.9328 in. From dimension H and the angle E the distance J is found to be 2.1638 in. From the dimension J and angle F, the distance L is then 2.1013 in. The radius R then is 1.3638 in. From the dimension J and the angle F the distance M is found to be 0.5314. Therefore the coordinants on the center of the gear are $X = 1.5314$, $Y = 2.2710$ in.

—GEORGE A. STROM,
Chicago, Ill.

We regret that space limitations will not permit publication of a number of equally interesting analytical solutions received in answer to Mr. Wiken's request. Although these solutions answer the request given in the problem, such solutions fail to satisfy the requirements of a gear drive. The design requires four gears having comparatively small numbers of teeth and pitch diameters proportional to the tooth numbers. Discussions of satisfactory gear designs for this problem will be given in the November number.

Fastenings

To the Editor:

Recommendations made by A. L. Foreman in the August *Product Engineering*, page 310, in particular as shown in Fig. 2B, are not very practical. To expand a $3\frac{1}{8}$-in. hole by heating until it passes over the 4-in. shaft end would require that the sleeve be heated to over 1,500 deg. F., only 100 deg. below the melting point.

If the taper is made less, it would not be very effective in preventing the parts moving endwise in relation to one another. To remove the sleeves when worn, even if the taper were reduced, would be just as expensive as drilling out the attaching screws. The sleeve could not be removed by expansion, as both parts would heat together, making it necessary to remove the sleeve by breaking or cutting it.

—FRANCIS W. SHAW
Heywood, Lancs., England

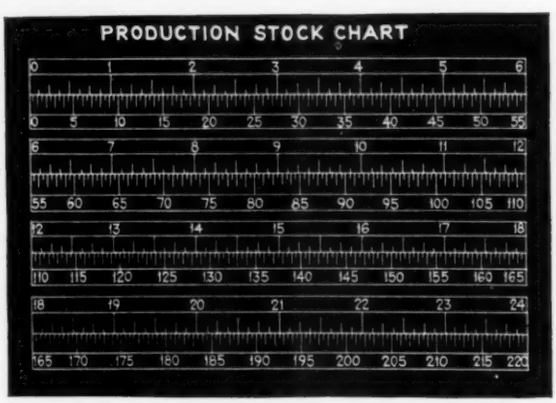
Correction

On the Reference Book Sheet, page 360 of the September number, in the equation for R the constant should be 11,260 instead of 1,260 as given.

Production Stock Chart

G. W. AUFMUTH
Cleveland, Ohio

A ready means for determining the length of stock required for a given number of pieecs of specified length is provided in the accompanying chart. Scales for the number of feet of stock needed for 100 pieces are placed directly beneath scales indicating the length of one piece. The upper scales include an allowance of 10 per cent for scrap. In addition to this allowance, the width of the cut-off tool, if one is used, must be added to the length of the part. As an example, for a lot of 100 screws $1\frac{1}{2}$-in. long the width of the cut-off tool, $\frac{1}{8}$ in., is added to the overall length, giving a total of $1\frac{5}{8}$ in. for one piece. From the $1\frac{5}{8}$-in. graduation on the upper scale, the lower scale indicates that 15 ft. of stock is required for 100 pieces. No cut-off allowance is needed for sheared stock, the 10 per cent scrap allowance being all that is required for sheared stock.



Upper scales denote inches of stock required for one piece. Lower scales denote feet of stock required for 100 pieces, including 10 per cent for waste

Art Versus Engineering Design

To the Editor:

In your May number there appeared a comment by George E. Hieber, who expressed himself in such a way as to lead one to believe that the present-day improvements are only in appearance and the so-called "guts" of the product is slighted.

There are quite a number of reasons for the application of art to products. Take the direct art appeal in sales, the product appealing in this way to the eye of the prospective buyer, maintaining its competitive sales value primarily by its artistry. The best engineering design in the world is largely wasted effort if it is unattractive to the buyer. A product which does not present an artistic touch is at a marked disadvantage in the present-day competitive field.

Appearance, as well as sturdiness and accuracy, have gone hand in hand through radio's shift to pressed-metal and die-cast construction. The mounted chassis is a compact unit, metal covered, shielded and sealed, fitting accurately into the base. The bulky, three-piece auto radio is now supplanted with a one-unit affair, combining a receiver, generator and speaker.

Bulky, unattractive parts are rapidly being replaced. Aluminum is taking the place of cast iron, a movement that has been accelerated by the demand for high speed. The airplane industry has unquestionably been the leader in pointing the way to lightness without loss of strength and rigidity. Strong aluminum alloys, magnesium alloys and the super-strong special alloy steels are rapidly coming to the forefront.

Appearance is improved by flange-mounted motors taking the place of foot-mounted motors. This also gives a compact arrangement, more rigid, a reduction in vibration, reduction in weight; and all at a lower cost.

All of the above have gone hand in hand with an improvement in appearance through the development of more pleasing lines and form. But practically every object in nature is endowed with color or creates a color sensation. Nature's multiplicity of colors is now being reproduced on man-made articles to make them more attractive and more salable.

Lacquer has opened new vistas to design and sales departments. Domestic machines, household equipment, and automobiles are now demanded in bright colors to match new tastes. And by the judicious use of color, a product can be made to look heavy or light, large or small, clean and cheery, or dark and depressing.

Use of color, finish and ornamentation in molded materials is a large factor. The field of application of molding materials is practically unlimited and the list is constantly growing. Many parts are now being made of molded plastics where formerly they were made of metal, wood or ceramics.

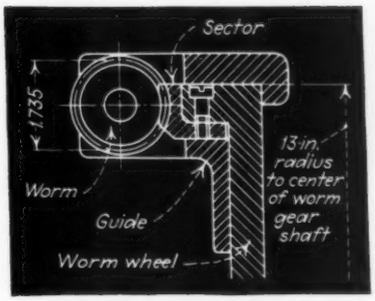
In this day and age of high competitive selling there must be something more than merely sturdiness and utility. A product that does not present an artistic touch is at a marked disadvantage in the present-day competitive field. —JOHN H. DAUM
Remington Rand Inc.

Lateral Worm

FRANK B. FAULKNER
Huntington, W. Va.

In the design of a special slotting machine, a self-locking ratchet-driven worm driving a sector was selected to swing a cutting tool through a small arc and lock the tool in place during the stroke. There was not sufficient headroom to place the worm in the usual position on the periphery of the sector, so the sector was designed with the worm at the side. The wormwheel then had a general appearance of a crown gear.

When the manufacture of this worm



A lateral worm with a wormwheel resembling a crown gear was used because there was no room for a conventional worm

and sector was placed in the hands of gear specialists, there was doubt as to the practicability of this layout. After some experimental work, however, the parts were made according to the design shown in the accompanying sketch.

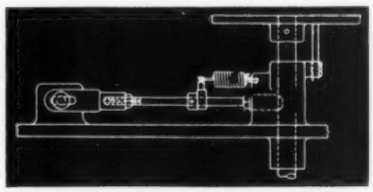
In cutting the sector the hob used was the same length as the worm, in this case 1-in. long. The sector teeth have a slight taper of about 1/32 in. similar to bevel gear teeth. The amount of this tooth taper will be inversely proportional to the length of the radius of the sector. Although in this case a radius of 13 in. was used, a radius as short as 6 in. would still provide a section strong enough for the work with a reasonable amount for wear.

Timed Transfer Action

F. SERVER
Springfield, Mass.

In an automatic machine a transfer shelf or plate, carrying a package was required to be lowered at the correct moment with the package held securely in place. No work other than holding the package and providing a transfer shelf across the gap was performed.

In the accompanying drawing, a cam below the lower extension of the shaft pushes the shaft up until a beveled pin snaps into a notch in the shaft, holding up the table while the cam goes down clear of the end of the shaft. A package is pushed on the



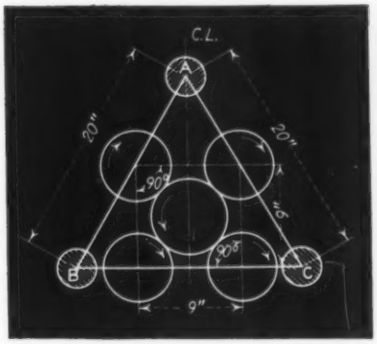
Beveled pin acts as a brake when the plate on this transfer mechanism is lowered

shelf by a mechanism from above, and pressure on the package causes the pin to ride out of the notch, pushing the shelf to a lower level. While the plate is traveling down, the beveled pin held by the tension spring acts as a brake, so that the package is held in place under constant pressure.

Another Problem

To the Editor:

Solutions submitted in the last number to Mr. Wiken's problem are interesting and informative, but how many shops could afford to find and compute these formulas? The radius and center of the mongrel gear can be developed by the shop in a comparatively short time. Turned blanks representing the three driven-gear pitch circles can be made and arranged as specified in the problem. From the center of an arc of any radius that is tangent to two of the pitch circles a line is drawn to a point on the line between the centers of the two pitch circles. This point is midway between the intersection of the two pitch circles. Three such lines are drawn, one for



The problem is to find the base of the isosceles triangle *ABC*

each pair of pitch circles, and the common intersection of the three lines is the center of the mongrel gear radius.

In a problem recently met in a specialty shop, although nearly all the dimensions were given, the shop had to develop some of the dimensions themselves. Four gears on 9-in. centers were driven by a common gear, all gears being the same size. Three guide pins formed an isosceles triangle, 20 in. on a side, with the base passing through two driven gear centers and each of the two equal sides passing through the other gear centers. As shown in the accompanying diagram the problem was simply to find the base of an isosceles triangle having two 20-in. sides circumscribing a 9-in. square. Can anyone supply these dimensions? —C. W. HINMAN
Orlando, Fla.

· NEWS ·

Streamlined Automobiles

Wind resistance of modern automobiles is about 20 per cent less than for cars of ten years ago, according to tests reported by R. H. Heald in Research Paper No. 591, published in the August *Bureau of Standards Journal of Research.*

As determined by wind tunnel tests the drag coefficient for a model of a present-day automobile is 0.0014 lb. for each sq.ft. of projected area, multiplied by the square of the speed in m.p.h. That still greater improvement is possible is shown by the coefficient of 0.0006 obtained with a model in which the body, rounded front and rear, inclosed the wheels, headlights and other projections. Complete streamlining using a contour resembling an airfoil with a flat lower surface gave a coefficient of 0.0005, only slightly lower.

The effect of cross wind, often felt as a sharp deceleration while driving, is said to be caused in part by the lateral component of the wind which causes tire deformation and increased tire resistance. Problems suggested by the investigation of the effect of cross-winds include that of maintaining steering control, and the determination of lift as well as drag and the moment about the center of gravity in the vertical plane.

High-Speed Multi-Section Car for Elevated Lines

Design features of the new light-weight, high-speed multiple-car for the New York Rapid Transit Corporation were announced recently. The car bodies, underframes and trucks, with the exception of wheels, axles, springs, wear plates and contact shoes will be entirely of aluminum. Weight has been reduced thereby from 210,000 lb. for the equivalent cars built of the usual materials, to 140,000 lb. estimated weight for the five-car articulated train.

Light-weight, streamlining and powerful motors with a new multi-notch automatic accelerating type of control contributed to increasing the rate of acceleration from 1¾ m.p.h. per sec. for cars of usual construction, to 3 m.p.h. per sec. for the new train when carrying the average load.

Other features of special interest are the use of twelve 60-hp. high-speed motors, two solidly mounted on each truck and connected to the axle through gear drive units and double flexible couplings; automatic compensation of the braking effort to compensate for the varying number of passengers on the train; and the streamlined shape. Balanced running speed on level tangent track at full normal voltage is 50 m.p.h. The air brakes are controlled electrically.

Permanent Magnets

Stabilizing treatments have been a problem of long standing in the selection and treatment of steel for permanent magnets in such electrical products as instruments, relays, magnetos and radios. Since Evershed's paper in 1920, but little has been published regarding the control of the loss of magnetic strength with passage of time.

At the National Metals Congress, a paper by R. L. Dowdell on "Investigation of the Treatment of Steel for Permanent Magnets—Part II," gives the results of tests conducted over a period of ten years. Materials tested cover a complete range of permanent magnet steels of 24 different compositions. A summary of the aging of these steels under seventeen different aging treatments indicates that both hardening and magnetic strains should be partially removed regardless of the composition.

Slow-Motion Movies Reveal Machine Defects

In the development of a high-speed addressing machine considerable trouble was experienced with the envelope in-feed operation. All the usual methods of investigation failed to reveal the defects that caused the envelopes to become jammed at the in-feed. Finally it was decided to make a micromotion study of the operation of the mechanism. In describing this investigation, R. Fawn Mitchell, manager of the technical department of the Bell & Howell Company, states:

"A micromotion outfit was arranged to take a close-up of the feeding mechanism with the feeding pawl painted white to facilitate following its motion. Motion pictures taken at 4,000 frames per min. disclosed that the feeding pawl vibrated at one time and not at another. Each time the pawl vibrated it failed to feed an envelope. Not only did the pictures show this effect, but they registered the time by means of a high-speed stop watch so that at least a reasonable approximation of the duration of the oscillation could be obtained. With this information the designers were able to effect the improvements immediately.

With increasing machine speeds it becomes difficult to analyze the actual machine operations. This undoubtedly accounts for the increasing use of micromotion pictures. But Mr. Mitchell cited another peculiar advantage that the designer gains by the use of motion pictures for the study of machine operations. Quoting Mr. Mitchell, "The motion picture camera is trained on just the essential field of action, and is viewed in a more or less darkened room free from distraction. As a result the attention is centered on the screen and the observer is not subjected to the many distractions of sight or sound which are a necessary concomitant to the actual machine. It is not always necessary to take slow movies to get results. Frequently ordinary speed movies will serve the purpose."

Opportunities for the Small Enterprise

Business enterprises have four distinct phases—the pioneer stage, the expansion period, the consolidation and stabilization phase, and, lastly, the period of disintegration, according to Crosby Field, vice-president of the Brillo Mfg. Company. Writing in the *Executives Service Bulletin* issued by the Metropolitan Life Insurance Company, Mr. Crosby states:

"Regardless of the cause, it is a fact that we found ourselves during the post-war period with a vast number of new needs. To fill these wants, new small companies started to manufacture. They grew very rapidly—some of them into quite sizable units almost overnight. We also improved our machinery for financing industrial enterprises. So, during the period 1926-1930, we had that ideal situation of a large number of companies just finishing their pioneer or expansion

phases and the necessary capital instantly available to consolidate them. In other words, to use a farm analogy, conditions were ripe for the harvest. And harvest them we did, but in harvesting we made no allowance for new sowing. Capital could not be attracted by the pioneer, inventor, or development engineer. Those with funds to lend asked themselves: Why take chances with new hazards when an investment in a large company would return the same yield with no apparent risk? So we ceased to sow. Not only did we cease to sow, but we completed our worship of size by solemnly announcing that the days of the garret inventor, the lone business man, or anything smacking of small size, were over.

"We must sow for awhile, building up new enterprises for perhaps another late harvest. But the main thing is to sow; to get back to work and study conditions. In so doing, we shall find many cases which present ideal opportunities for small enterprises."

Mr. Field did not state specifically that his philosophy applies to both small and large business concerns, but his discussion leads one to believe that he considers that it applies equally well to both.

New Name for the A.S.S.T.

Recognizing that the modern activity of the organization has outgrown its present name, the new name, "American Metals Society," has been selected by the board of directors of the American Society for Steel Treating. For several years the principle activity of the A.S.S.T. was to disseminate information on the heat-treatment of tool steels and alloy steels. But with the advent of the newer alloy steels and the widened interest in fine metals more attention was given to alloying, manufacturing and mechanical treatment. This change was reflected in the society programs, where an increasing amount of time was devoted to subjects other than steel treating. Hence, the selection of the new name for the society.

The Engineer's New Task

What is the field for the engineer under the new industrial system? Although books have been written on the subject, the situation is summarized clearly in a short statement by Maurice Holland, director of the division of engineering and industrial research of the National Research Council. Pointing out that under present conditions only a negligible part of the new products developed in research laboratories are being integrated into the industrial structure, Mr. Holland states that the engineer will serve as the necessary link between the inventor and the industrial backer. In outlining the new field for the engineer, Mr. Holland states:

"In the past the engineer's talents have been devoted to increasing productive efficiency and cutting down operating expenses in existing industries. His work was confined to perfecting machinery and processes so that a greater output of an established product would be possible at less expense for labor and other production costs.

"In the last few months the industrial demands upon the engineer have changed. The need today is for new industries to employ more workers and for the development, in existing industries, of new or related articles which can be marketed in addition to established products.

"The engineer fits into this new requirement in an ideal manner. His training is such that he has a basic grounding in science and technology which gives him the ability to appraise the possibilities of the products and processes developed by the inventor and the research laboratory. He also has enough training in business and economics to enable him to present the potentialities of a new product in such a way as to enlist the support of financial backers."

To illustrate the extent of this new field, Mr. Holland points out that research carried on in the last year and a half has developed more than a score of new products and processes in the food industry alone. In the chemical industry new products such as fertilizers, synthesis of petroleum products, new resins and solvents, liquid oxygen explosives, new alloys, and new dyestuffs await exploitation.

All of them are products which can be introduced as sub-industries and which may in time grow as did the radio, plastics and rayon industries, from small off-shoots of an established business into major industrial enterprises.

MEETINGS

National Metals Congress—Fifteenth annual congress, Oct. 2-6, Detroit, Mich. Sponsored by American Society for Steel Treating, W. H. Eisenmann, secretary, 7016 Euclid Ave., Cleveland, Ohio.

American Welding Society — Annual fall meeting, Oct. 2-6, Book-Cadillac Hotel, Detroit, Mich. Miss M. M. Kelly, secretary, 29 West 39th St., New York, N. Y.

American Institute of Mining and Metallurgical Engineers — Institute of Metals, and Iron and Steel Division, Hotel Statler, Detroit, Mich., Oct. 2-7. A. B. Parsons, secretary, 29 West 39th St., New York, N. Y.

Association of Iron and Steel Electrical Engineers—Twenty-ninth convention, Oct. 17-19, William Penn Hotel, Pittsburgh, Pa. John F. Kelly, managing director, Empire Bldg., Pittsburgh, Pa.

EXHIBITIONS

National Metals Exposition—Oct. 2-6, Detroit, Mich. W. H. Eisenman, secretary, 7016 Euclid Ave., Cleveland, Ohio.

American Bottlers of Carbonated Beverages — Annual exposition of machinery and equipment, Oct. 9-13, Jefferson County Armory, Louisville, Ky. James Verner, Jr., chairman, 237-239 Woodward Ave., Detroit, Mich.

National Business Show—Grand Central Palace, New York, N. Y., Oct. 16-21. Edwin O. Tupper, secretary, 50 Church St., New York, N. Y.

Iron & Steel Exposition — Oct. 17-19, William Penn Hotel, Pittsburgh, Pa. John F. Kelly, managing director, Empire Bldg., Pittsburgh, Pa.

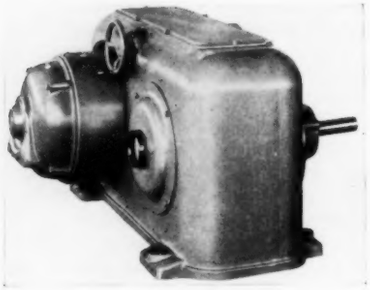
Postgraduate Course in Kinematics

Recent development in the theories of the kinematics of machinery now make it possible to solve on the drawing board problems whose solutions formerly required experimental models. This has been accomplished through the extension of the theory of equivalent mechanisms. As an example, the application of this theory makes it frequently possible to substitute link mechanisms for expensive cams.

To make available to designers a working knowledge of these advanced theories of kinematics, the Polytechnic Institute of Brooklyn has established a postgraduate evening course, "Kinematics of Machine Design," the first semester of which started on September 28. The lecturer is Mr. R. deJonge, Dipl.-Ing., who has long been associated with this work.

NEW MATERIALS and PARTS

Sterling Vari-Speed Motor



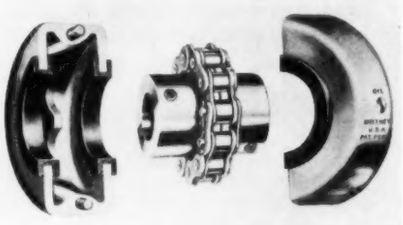
Incorporates in a single unit an electric motor, speed reducer with infinite variation in ratio, and a speed indicator. A flexible V-belt running on two V-pulleys mounted on parallel splined shafts are actuated simultaneously by a small handwheel that changes the diameters of both pulleys and maintains the belt tension. Available in single or double reduction giving speeds as low at 5 r.p.m., or with increasers giving speeds up to 15,000 r.p.m. Speed variation of approximately 4 to 1 is provided by the adjustment. Electrical remote control box with an electrical speed indicator also available. Sterling Electric Motors, Inc., Los Angeles, Calif.

Indicating Temperature Controllers

For air, gases, or liquids in ducts, ovens, tanks, vats, pipes, are self-contained and actuate electric contacts. Cover has a window for viewing the indicator and setting mechanism. Temperature sensitive element extends from the back of the case on an extension long enough to project through walls up to 5-in. thick. For use with liquids the sensitive element is inclosed in a well. Expansion and contraction of a bimetal helix operates the contact mechanisms. Type L-48 is a two-wire line voltage control which closes a circuit on temperature rise, incorporates a single circuit mercury switch. Operating differential said to be approximately 5 deg. F.; electrical rating, 10 amp. at 110 volts, 5 amp. at 220 volts, a.c. or d.c. Type L-68 for 3-wire line voltage control incorporates a double circuit mercury switch, rated at 3 amp. 20 volts, 1 amp. 110 volts, ½ amp. 220 volts, a.c. or d.c. Operating differential approximately 3 deg. F. Type L-93 for use with proportioning motors on a.c. current only, has modulating range of 1 deg. F. Standard scale ranges are 0 to 200 deg. F., and 150 to 400 deg. F., special ranges available. Approximate dimensions are $5\frac{5}{8}$-in. high, $4\frac{3}{16}$-in. wide, $8\frac{7}{8}$-in. long. Minneapolis-Honeywell Regulator Co., Minneapolis, Minn.

Whitney Flexible Coupling

Consists of two accurately cut roller chain sprockets coupled together with roller chain. Clearance between the two sprockets is designed to allow for shaft misalignment. Said to be quickly installed or dismantled. Supplied with



a cover designed to protect the coupling from dirt, and to retain lubricant. Whitney Mfg. Co., Hartford, Conn.

Weston Photo-electric Kit

Containing all equipment necessary for photo-electric controls such as a smoke detector, burglar alarm, turbidity detector, door opener, counting device, safety device. Photocell included in the kit is the photronic cell, a photo-voltaic type which operates relays directly without the use of auxiliary apparatus or batteries. Also included are connection diagrams for typical applications. Kit is assembled especially for those preferring to do experimental work in connection with the control of mechanical equipment or processes. Weston Electrical Instrument Corp., 614 Frelinghuysen Ave., Newark, N. J.

Carter Genemotor

Operating from a direct-current source, provides a source of a.c. voltage for electronic devices. Special circuits built into the motor are said to increase the efficiency, and to provide better regulation and higher power output. Completely inclosed unit is equipped with bearings requiring no oiling during the life of the machine. Rubber cushions between the base and frame are provided to absorb vibration. Available for 6 or 32-volt d.c. supply, and in two capacities, 135 and 180 volts. Requires a mounting space of $4\frac{1}{4}$x$4\frac{1}{4}$ in., weighs 7 lb. Carter Genemotor Corp., 361-365 W. Superior St., Chicago, Ill.

Taylor Pipe Fittings

Fabricated from 0.134-in. steel by welding, are furnished plain end, flanged both ends, or flanged one end only, with flanges of forged steel. Center to end dimensions conform with 125-lb. American Standard. Elbows available in nine sizes from 3-in. to 16-in. inside diameter, designed for hydraulic pressures up to 200 lb. per sq.in. Welded steel construction is said to have advantages over cast iron in smoothness, strength and weight reduction. Compared with cast iron these fittings are said to reduce weight

by more than one-half, resulting in lower erection costs and lighter supports. Fittings are also said to resist shock and rough handling, to be easily welded in the line. All fittings are tested to a pressure 50 per cent greater than the working pressure. Taylor Forge & Pipe Works, P. O. Box 485, Chicago, Ill.

General Electric Utility Timer

Driven by a Telechron clock motor, turns an electric circuit both on and off once every 24 hours. Furnished in a die-cast case with a clock face and setting dials. Also available as a thermostat timer for use with double range thermostats for control of house heating systems. This type is mounted in a rust-resisting case of drawn steel with an aluminum finish. Five knockouts are provided in the case for connections. Clock carries an a.m. and p.m. indicator and a dial for setting the clock at a.m. or p.m. time. Two other pointers are provided to set the mechanism to the proper time for thermostat operation. General Electric Co., Schenectady, N. Y.

Satco Bearing Metal

Is said to combine the hardness of more expensive bearing metal compositions with the advantage of an initial melting point 100 deg. F. higher. Recommended for use in railway work as a lining for truck and trailer brasses, car journal bearings, lateral plates for engine trucks, trailer and driving boxes, crosshead gibs; also for internal combustion engine bearings where it is claimed to be superior to high tin babbitt metals. Is said to be as easily handled in casting as any of the regular babbitt metals and to adhere to bronze and steel-backed bearings. Lower shrinkage is said to be of assistance in die-casting this material. National Lead Co., 111 Broadway, New York, N. Y.

Signal Junior Relay

Identified as Type C-7, relay is mounted on a bakelite base $2\frac{1}{2}$x$2\frac{3}{4}$ in., arranged for back-connected switchboard mounting or front-of-panel mounting as illustrated. When relay is energized, each contact arm completes electrical circuit through two stationary contacts, no pig-tail connections being used. Electromagnet construction said to eliminate chattering. Contacts can be arranged to provide a variety of connections such as two breaks on each side of a circuit,

one break in each of two independent circuits, four breaks in series or in parallel in a single circuit. Contacts rated at 15 amp., 110 volt, a.c. Signal Engineering & Mfg. Co., 162 W. 14th St., New York, N. Y.

Brown & Sharpe Steel Rule

To facilitate measurements in decimals, is graduated in 10ths and 50ths on one side, the 50ths being figured every 10th. Hundredths of an inch can be estimated readily. Opposite side of the rule is graduated in 32nds and 64ths, the 64th graduations being figured every 8th in. Rule No. 315 with these graduations is 6-in. long. Available in a dull finish aluminum holder with decimal equivalents on one side. Brown & Sharpe Mfg. Co., Providence, R. I.

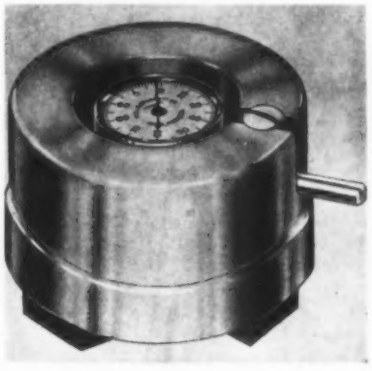
General Radio Adjustable Voltage Transformer

For illumination control, speed control on small motors, testing of household appliances, laboratory source of adjustable voltage. Available in a protecting case with an attachment cord and an outlet receptacle. Maximum current rating 5 amp., with a continuous voltage adjustment between 0 and 130 volts when operated from a 115-volt, 60-cycle line. Said to incorporate smooth control of a rheostat with the high efficiency of a trans-

former. Contact mechanism designed to eliminate short-circuited turns. General Radio Co., Cambridge, Mass.

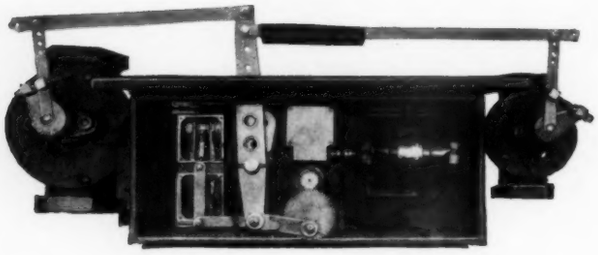
Starrett Vibrometer

Measures the amplitude of vibration in high-speed rotating machinery. It is said to be particularly intended for comparative readings. Consists of a dial indicator set in a heavy metal retaining ring. Three soft rubber shoes held in grooves in the retaining ring conform to the contour or incline of the machine surface and prevent slipping. Designed so that the heavy retaining ring, because of its inertia, tends to remain stationary in space, permitting vibration of the shaft to operate the hand of the indicator dial. A sensitivity adjustment is provided and the dial gage is removable. All parts are chromium and nickel plated. Supplied with a case. L. S. Starrett Co., Athol, Mass.



Eby Trimming Condensers

Available in single or double assembly, having $\frac{1}{2}$-in. mounting centers and $1\frac{1}{16}$-in. diameter base. Base is made of ceramic material with a power factor of 0.4 to 0.5 per cent,



has low porosity, high dielectric strength. Plates are of tinned spring bronze. Both assemblies available in a low range of capacities from 10 to 80 Mmf.; a high range of capacities from 700 to 1,000 Mmf. on the single and up to 800 Mmf. on the double also available. Adjustment for either type is made from one side of the condenser. Designed to prevent shock and vibration from varying capacity. Capacity can be tapped one or two places at any portion of the maximum range. The H. H. Eby Manufacturing Co., 21st St. & Hunting Park Ave., Philadelphia, Pa.

Micarta for Roll-Neck Bearings

Available in two grades: No. 1C17, for the radial part of roll-neck bearings on severe applications, and No. 863 for thrust plates; are green in color to distinguish them from No. 1B21 and No. 262 used on lighter pressure applications. Materials can be machined or sawed to desired shape. Claimed to stand higher thrust loads and pressures without dislamination, splitting, or working out of the chuck. Westinghouse Electric & Mfg. Co., East Pittsburgh, Pa.

Multi-Position Temperature Controller

For applications where the heat demand is subject to wide variations over varying periods, such as a heat-treating furnace in a production line subject to changes in rate of flow. Type 305 controller is actuated by an instrument having five contacts, two on either side of the normal setting. Through a limit switch arrangement, each intermediate contact has two valve positions selected according to the direction of change. Controller is said to be independent of time lag and to provide close control without hunting. For application between the two- or three-position controller and the complete balancing controller. Automatic Temperature Control Co., Philadelphia, Pa.

NEW BOOKS and PUBLICATIONS

New Edition of the S.A.E. Handbook

S.A.E. Handbook, 1933. 733 pages, 5¼x8¼ in. Fabrikoid covers. Published by the Society of Automotive Engineers, 29 West 39th St., New York, N. Y. Price $5.

In this edition specifications are grouped according to type rather than the particular kind of automotive apparatus to which the specifications apply. Titles of the nine sections are: Unit Parts and Fittings, Process Materials, Fabricated Materials, Screws and Washers, Tests, Transportation and Maintenance, Tools and Production Equipment, Nomenclature and Definitions, Miscellaneous. A final section includes Standards committee regulations, a list of A.S.A. standards, S.A.E. Standards committee personnel and a general index.

Three pages are devoted to inch-millimeter conversion as approved by the A.S.A. in 1933. Omitted in this edition are forty-three obsolete specifications; among the revised specifications are those on propeller shaft couplings, ball and roller bearings, poppet valves, wrought aluminum and aluminum bronze, cast brass and bronze, wrought copper and copper alloys, insulated cable, automobile wiring, screw and bolt heads and nuts. The larger page size with larger type will be well received.

Handbook of Mathematical Tables and Formulas

Dr. Richard S. Burington. 264 pages, 5x8 in. 57 figures. Flexible fabrikoid covers. Published by Handbook Publishers, Inc., Sandusky, Ohio. Price $2.

Engineers will welcome this work, combining essential formulas and tables in one convenient volume. Well indexed, carefully printed, and conveniently bound, it shows uncommon attention to the needs of a table-user. It is in two parts, the first containing formulas and theorems from elementary mathematics, and the second part including logarithms, constants, trigonometric functions, powers, roots, reciprocals, probability and other useful tables.

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Conveyor—Link-Belt Co., 2410 W. 18th St., Chicago, Ill. Data Book No. 1289, 128 pages, "Caldwell Helicoid and Sectional Flight Conveyor." Well-illustrated introduction includes typical applications; data on materials handled, capacity charts, power requirements, layout and installation. Catalog information on conveyors and fittings, and a section on selection of drives, speed reducers, motors.

Cork—Mundet Cork Corp., 450 Seventh Ave., New York, N. Y. Booklet, 20 pages, "Jointite Cork Products," with an introduction on the history and manufacture of cork. Details on general cold storage insulation using corkboard and molded cork pipe covering. Data on roof insulation, heat-loss charts, and information on vibration isolation.

Relays—Ward Leonard Electric Co., Mt. Vernon, N. Y. Bulletin No. 106, catalog page, "Midget Magnetic Relays," includes a number of wiring diagrams, contact ratings.

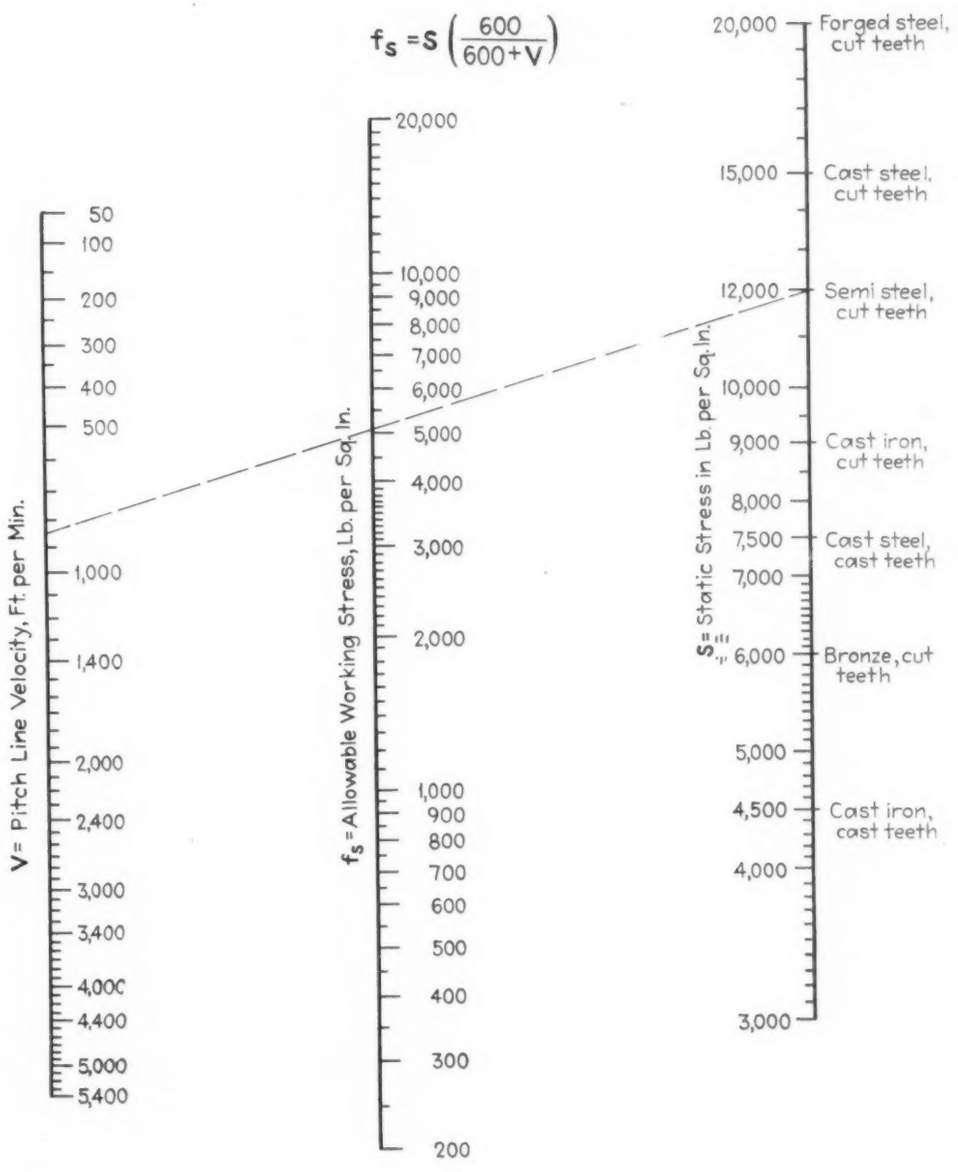
Solenoid Valves—W. H. Nicholson & Co., Wilkes-Barre, Pa. Bulletin 633, four pages, "Nicholson Solenoid-Operated Valves," with data on operation, construction and current consumption.

Valves—Jenkins Bros., 80 White St., New York, N. Y. Catalog No. 23, 263 pages, "Jenkins Valves and Mechanical Rubber Goods." An introduction on service facilities includes six full-page etchings of operations in the manufacture of valves. Catalog chapters on bronze valves, iron and steel valves, miscellaneous equipment, mechanical rubber goods, gage glasses. Useful engineering data included in a 33-page section covering standard flange dimensions, pipe dimensions, properties of saturated steam, flow of steam through pipes and orifices, and other helpful tables.

Welded Pipe—The Linde Air Products Co., New York, N. Y. Handbook, 156 pages, "Fabrication of Oxwelded Piping," giving data and procedure for fabrication and erection of piping systems. Separate divisions on fabrication of steel and wrought-iron piping, templet layout for welded fittings, tables for estimating costs, welding of cast-iron and non-ferrous piping. Introducing the first part is an 8-page section on the fundamentals of pipe welding. Procedure control, layout and fabrication, welding practices, qualification tests for welders also included in the first section. Well illustrated with many useful tables.

Allowable Unit Stress of Different Gear Materials at Various Velocities

Contributed by E. Cowan
New York, N. Y.



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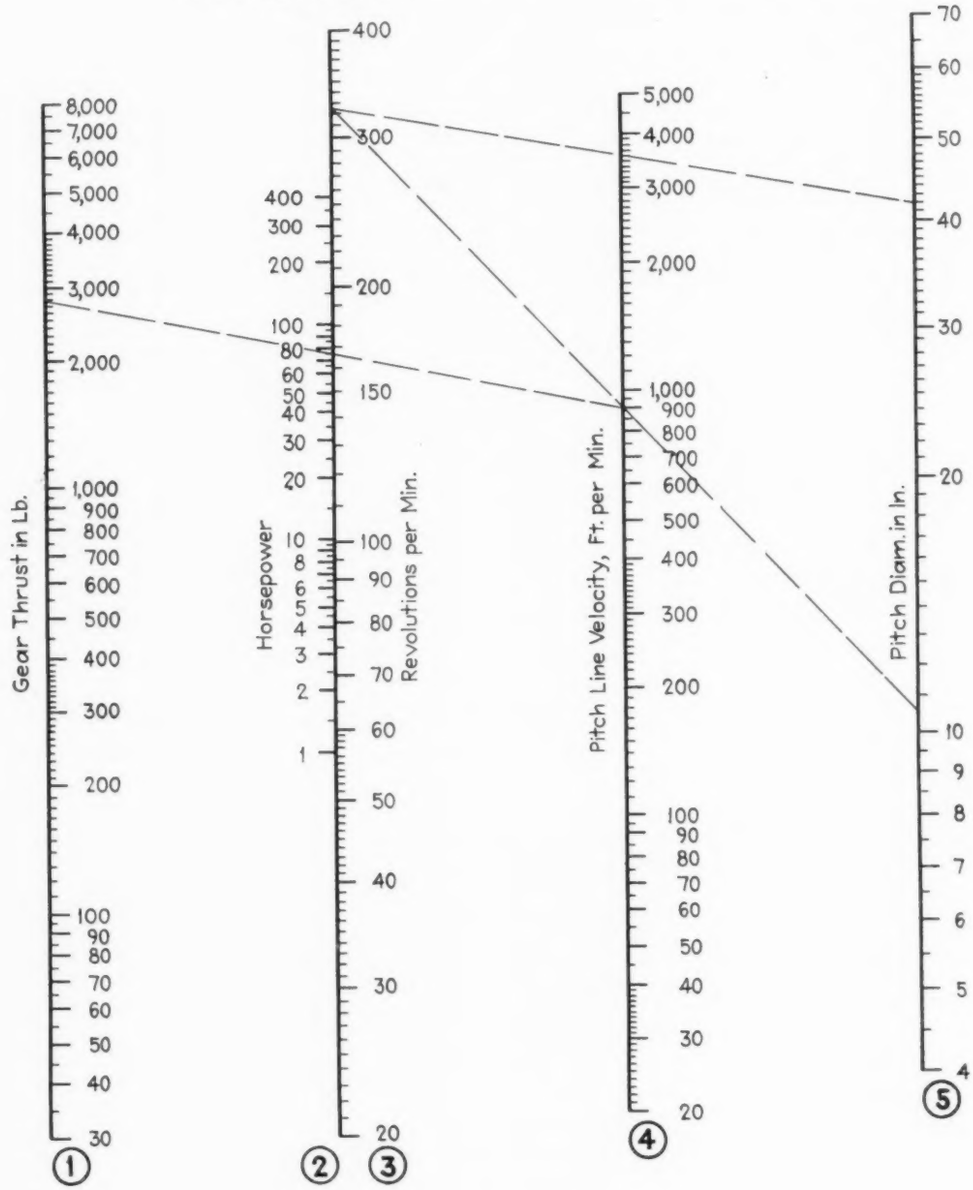
Gear Thrust for 14½-Deg. Spur Gears

For angles of action other than 14½ deg., divide the gear thrust as found by 0.25 and multiply by the sine of the angle of action.



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